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## U. S. S. *TEXAS*.

### DESCRIPTION AND OFFICIAL TRIALS.


BY HENDERSON B. GREGORY, ASSOCIATE.


Battleship No. 35, the *Texas*, is a sister-ship of the *New York*, now nearing completion at the Navy Yard, Brooklyn, New York. Both vessels were authorized by an Act of Congress approved June 24, 1910, which provided for the construction of two first-class battleships.

The *Texas* is a twin-screw vessel fitted with reciprocating engines, and designed for a speed of 21 knots, at 27,000 tons displacement, with the main engines developing 28,100 I.H.P. She was built under contract by the Newport News Shipbuilding and Dry Dock Co., of Newport News, Virginia. The contract was signed December 17, 1910, the price being $5,830,000.00 and time of construction 36 months.

## PRINCIPAL HULL DIMENSIONS.

| | |
|---|---|
| Length between perpendiculars, feet and inches | 565–00 |
| on L.W.L., feet and inches | 565–07¼ |
| over all, feet and inches | 573–00¾ |
| Breadth, extreme, on L.W.L., feet and inches | 95–02⅝ |
| molded, feet and inches | 94–10⅛ |
| Depth molded, main deck at side M.S., feet and inches | 48–08 |
| Draught to L.W.L., feet and inches | 28–06 |
| Displacement corresponding, tons | 27,000 |
| Ratio of length to beam | 5.934 |
| Displacement per inch at L.W.L., tons | 91.75 |
| Area of midship section, square feet | 2,652 |
| L.W.L. plane, square feet | 38,560 |
| wetted surface, square feet | 65,700 |
| Coefficient of fineness, block | 0.6163 |
| midship section | 0.9773 |
| L.W.L. plane | 0.7167 |
| cylindrical | 0.6307 |

## GENERAL DESCRIPTION OF HULL.

The hull is of steel throughout and of typical American design, with all turrets on the center line, two smoke pipes and the usual cage masts.

*Masts.*—There are two cage masts, 124 feet high above the load water line, located at frames Nos. 51½ and 76½; on which are located the spotter's tops, upper and lower searchlight platforms—mounting one light each; wireless, signal yards, etc. The forward mast also contains the conning platform and the after one the torpedo-defense station.

*Bridge.*—The bridge is just below the conning platform. It covers the bridge deck and has portable extensions at either side, reaching to the vessel's sides. The conning tower is located at the forward end of the bridge.

*Bridge Deck.*—This is a small deck at the base of the forward mast, extending from frames Nos. 44½ to 54. It contains the chart house, and its forward end is taken up with the conning-tower foundations.

*Superstructure Deck.*—This deck extends from the after side of turret No. 2 to frame No. 54, and contains a deck house in which are the captain's quarters.

*Main Deck.*—The main deck, on which is located the main battery, described elsewhere, is a weather deck throughout. Forward are located the windlass and a deck winch, and far aft a capstan. There are four deck houses, as follows: One forward, in which are the admiral's and chief of staff's quarters, the captain's and fleet officers' offices. In another, at the base of the forward smoke pipe, are the officers' galley, bakery, lamp room and lockers. A third, at the base of the after smoke pipe, contains the blacksmith shop, foundry, butcher shop and lockers. The fourth, below the after mast, is exclusively for the crew's galley. The two boat cranes, each mounting a searchlight platform for two searchlights, are located amidship, port and starboard.

*Gun Deck.*—The gun deck extends from the stem to the stern. Forward of the diagonal armor are the wardroom officers' quarters. Within the casemate is part of the 5-inch battery, described elsewhere; printing office, ordnance stores and issuing room, wardroom officers' bath and water closets, foremen's and mess attendants' wash rooms, drying rooms, wireless, coaling engines and bread rooms, general mess condiment, issuing room, lucky bag and post office. Aft of the diagonal armor is the crew's space, executive, ordnance and engineer officers' offices, paymaster's office, ship's stores, surgeon's examining room, dispensary, operating room and sick bay, laundry, master-at-arms and sergeant of marines' staterooms, general mess pantry, armory, chief petty officers' and crew's washrooms and water closets, and electric capstan motor.

*Half Deck.*—This deck is forward of the forward diagonal armor, between the gun and protective decks. On it are located the junior and warrant officers' quarters, starboard and port respectively, and cleaning-gear room in the peak. Just within the diagonal armor are two blower rooms at about this deck level.

*Protective and Berth Decks.*—The protective deck extends from the bow to frame No. 137, sloping off aft from frame No. 122. From the latter frame the berth deck, containing

the chief petty officers' and crew's spaces, is carried aft at the protective-deck level. On the protective deck, from forward-aft, are located the paint rooms, chain lockers, forward ice-machine room, stores, blower rooms, crew's space, central station, band room, storage-battery charging station, coal bunkers, ammunition passages, boiler hatches, evaporator room, blower rooms, prison, clothing and small stores, crew's space, general workshop, ice machine and refrigerating rooms.

*Upper Platform.*—Next below the protective deck is the upper platform. Forward of the boiler rooms are electrical stores, distribution-board room, sub-central, intercommunication room, magazines, handling and shell rooms, windlass machinery, stores, pump room, chain lockers, paint and oil rooms. Extending through the boiler rooms, on the center line, is the wiring passage. The forced-draft blower rooms are located at this level in each fireroom, with coal bunkers outboard. Between the boiler and engine rooms is another distribution-board room; magazines, handling and shell rooms, with steam-pipe passages outboard. Abreast of the engine rooms are blower and store rooms. Abaft of the engine rooms are magazines, handling and shell rooms.

*Lower Platform.*—Like the former, the lower platform is interrupted by the machinery spaces. Forward are the forward trimming tanks, sails and awnings, hold, navigator's and ordnance stores, magazines, handling and shell rooms, torpedo room and forward dynamo room. Coal bunkers extend abreast of the boiler rooms, port and starboard, and the after dynamo room, together with magazines, handling and shell rooms are between the boiler and engine rooms. Again store rooms outboard of the engine rooms; and store rooms, steering-engine room, magazines, handling and shell rooms, pump and steering-gear rooms aft.

*Hold.*—In the hold, from forward-aft, are located the forward trimming tanks, fresh-water tanks, stores, forward dynamo-condenser room, boiler rooms, with coal bunkers outboard, after dynamo-condenser room, provisions, engine

rooms, with wiring passage between; engineer's, marines' and ordnance stores, and the after trimming tanks.

*Double Bottoms.*—The double bottoms extend from frames Nos. 9 to 122. That portion under the boiler rooms, frames Nos. 61 to 73, forms the reserve feed-water tanks. Fuel oil is carried in tanks under the machinery spaces, frames Nos. 79 to 99. All other double bottoms are dry.

*Capacities of Double-Bottom Compartments.*

| Com'p't. | Fresh water, tons. | Salt water, tons. | Com'p't. | Fresh water, tons. | Salt water, tons. |
|---|---|---|---|---|---|
| A–92 | 57.78 | 59.51 | B–89 | 67.22 | 69.24 |
| A–93 | 55.05 | 56.70 | B–90 | 67.22 | 69.24 |
| A–94 | 70.19 | 72.29 | B–91 | 67.14 | 69.15 |
| A–95 | 70.19 | 72.29 | B–98 | 67.19 | 69.21 |
| A–96 | 104.83 | 107.97 | B–99 | 67.19 | 69.21 |
| A–97 | 95.50 | 98.37 | C–98 | 60.67 | 62.49 |
| A–98 | 65.56 | 67.53 | C–99 | 60.69 | 62.50 |
| A–99 | 99.89 | 102.89 | D–97 | 90.61 | 93.33 |
| B–86 | 52.83 | 54.41 | D–98 | 94.24 | 97.07 |
| B–87 | 52.89 | 54.48 | D–99 | 110.25 | 113.56 |
| B–88 | 67.22 | 69.24 | | | |

## COMPLEMENT.

The ship's complement will be approximately as follows:

| | |
|---|---|
| Flag Officer | 1 |
| Chief of Staff | 1 |
| Commanding Officer | 1 |
| Wardroom Officers | 30 |
| Junior Officers | 18 |
| Warrant Officers | 12 |
| Crew (including 75 marines) | 1,009 |
| Total | 1,072 |

## BATTERY.

There are ten 14-inch guns in the main battery, arranged in pairs, in five turrets on the main deck along the center line of the vessel. Turrets Nos. 1 and 2 are forward, the latter being placed at sufficiently high elevation to permit ahead fire over the top of the former. Turret No. 3 is between the after mast and the engine hatches, and Nos. 4 and 5 are grouped abaft the engine hatches, the former elevated for astern fire over the latter.

The turrets are electrically operated, each containing the following electric apparatus:

| No. | Character of Apparatus (Diehl Motors). | H.P. each. |
|---|---|---|
| 2 | Training motors | 25 |
| 2 | Upper ammunition hoists | 40 |
| 2 | Lower ammunition hoists | 10 |
| 2 | Elevating motors | 15 |
| 2 | Rammers | 10 |
| 2 | Ventilating blowers | 4.4 |

A secondary battery, of twenty-one 5-inch rapid-fire guns for torpedo defense, is also provided. Nineteen of these guns are distributed along the gun deck as follows: Divided port and starboard—four in the officers' quarters forward, ten in the casements and four in the crew's space aft, with one at the extreme stern on the center line. The two remaining guns are mounted on the bridge deck, port and starboard.

There are twenty-five electric-chain ammunition hoists for the secondary battery, each driven by a 3-horsepower Diehl motor.

The following smaller guns are also provided:

4 3-pdr. guns for saluting;
2 1-pdr. guns for boats;
2 3-inch field pieces;
2 0.30-caliber machine guns.

There are also four 5-m. by 21-inch submerged torpedo tubes forward. For changing the torpedoes, two Mark XIV, Ingersol-Rand, electric-driven air compressors are provided. Each has a capacity of 30 cubic feet of air per minute, at 2,500 pounds pressure per square inch. The driving motors are 50 horsepower each. The plant is provided with two accumulators of 25 cubic feet capacity each.

### SMALL BOATS CARRIED.

The following small boats are carried on the main deck and in skid deck beams at the superstructure-deck level abeam of the smoke pipes:

2 50-foot steamers;
2 40-foot steamers;
2 40-foot motor sailing launches;
2 36-foot sailing launches;
1 31-foot racing cutter;
2 30-foot cutters;
2 30-foot metal whale boats;
2 20-foot dinghies;
2 14-foot punts.

Two electrically-operated boat cranes are provided for handling the boats, except the whale boats, which are hung in davits aft. Each crane has two operating gears—one for turning and the other for hoisting. The operating gears are driven by Diehl motors of 40 and 50 horsepower, respectively.

### ANCHOR WINDLASS.

The anchor engine is located on the upper platform, frames Nos. 18 to 24. The engine is of the horizontal, double-cylinder, reversible type, built by the Hyde Windlass Co., of Bath, Maine. The cylinders are 17 inches diameter each by 14 inches stroke.

The windlass is driven by a worm gearing direct from a worm on the engine crank shaft. Attached to the worm gear-

ing are two vertical shafts, each fitted on its upper end, at the main deck, with a wildcat, and so arranged that the wildcats can be operated together or independently of each other.

### DECK WINCH.

An electrically-operated deck winch is located on the main deck, frames Nos. 22–24, center line. It is compound geared and has a lifting capacity of 16,000 pounds at 50 feet per minute, or 4,000 pounds at 250 feet per minute. The winch was made by the Hyde Windlass Co., of Bath, Maine, and is operated by a 35-horsepower Diehl motor.

### COALING ENGINES AND GEAR.

There are two steam-driven coaling engines made by the Hyde Windlass Co., of Bath, Maine. They are located amidship on the gun deck, frames Nos. 64 to 66. The engines are of the vertical, double-cylinder type, with cylinders 14 inches diameter by 10 inches stroke.

Each engine, through miter gears, drives an athwartship shaft, which in turn drives, port and starboard, fore-and-aft shafts; to which are geared eight gypsy heads each, which can be thrown in or out of gear at will by friction clutches. The athwartship shafts are cross-connected so that either engine can operate all gypsy heads if necessary. The capacity of the gear is 2,400 pounds on each gypsy head, at 200 feet per minute.

### CAPSTAN.

Located at frames Nos. 128–129, center line, on the main deck, is an electric capstan, made by the Hyde Windlass Co., of Bath, Maine. It is compound geared, of same capacity as the deck winch and operated by a similar motor, located below on the gun deck.

### STEERING ENGINE AND GEAR.

The steering-engine room is on the starboard side of the center line, just abaft of the engine rooms, access being from the starboard engine room. The engine, of the Hyde Wind-

less Company's vertical, double-cylinder type, has cylinders 18 inches in diameter by 14 inches stroke.

From the steering engine a shaft is led aft to the tiller room, where is located the main steering gear, consisting of a right-and-left-hand screw, on which are two driving nuts direct connected by side rods to the rudder-stock crosshead. The screw is operated through gearing by the shaft, or emergency hand-steering gear, either of which may be disconnected when not in use.

In addition to the steam-steering engine, an electric gear is installed, driven by a 150-horsepower Diehl motor. It is located in the steering room and operates the steering gear through the engine shaft, suitable clutches being provided for throwing the engine and the motor in or out of gear.

The steering engine is controlled by telemotor, fitted in duplicate, from the steering platform, conning tower and central station. It may also be direct controlled by a handwheel in the steering room aft and one at the engine.

The control of the electric gear is from the same stations, steering-engine room excepted, as the steam gear.

The usual emergency hand gear is located in the steering room. It consists of four large hand-steering wheels mounted on a common shaft, which is connected through a train of gears and suitable cutout clutches, with the steering gear.

## LAUNDRY.

A well equipped laundry, with capacity to wash for about 100 men, is located on the gun deck, frames Nos. 106 to 114, port side.

The laundry machinery is driven by a 6-horsepower Diehl electric motor, through suitable shafting and belting, and comprises the following apparatus:

1 Washer;
2 Wash tubs;
1 Soap tank (90 gallons);
1 Starch kettle (10 gallons);

1 Extractor;
1 Combination ironer;
1 Band ironer;
1 Steam-heated bosom press;
1 Reverse body ironer;
1 Ironing table;
1 Drying room.

## GALLEYS AND BAKERY.

The galleys and bakery are located in deck houses on the main deck, and, except for the customary steam kettles and coffee urns, are electric throughout; no coal ranges nor bake ovens being used. The officers' galley contains a five-section electric range, and that for the crew one of ten sections. The ranges were supplied by the General Electric Co.

Located in the crew's galley are the following electrically operated machines:

| No. | Character of Machine. | H.P. of Motor. |
|---|---|---|
| 1 | Potato peeler ..... | 1 |
| 1 | Meat chopper ..... | ¾ |
| 1 | Cake mixer ..... | 1 |

The bakery is equipped with two electric bake ovens and one dough mixer driven by a 2-horsepower motor.

An electrically-driven dish-washing machine, with 2-horsepower motor, is provided for the general mess pantry.

## DRAINAGE SYSTEM.

*Main Drain.*—The main drain, 15½ inches inside diameter, runs from the forward fireroom, in a single pipe, port side, to the forward bulkhead of the engine room, where it branches into two full-size pipes—one continuing aft through the port engine room, and the other athwartship, thence aft through the starboard engine room, both connecting to flanges provided on the main circulating-pump suction pipes.

There is a 15½-inch stop-check valve in the system in each

engine and boiler compartment, for draining same, operated at place and from the protective deck. There is also a stop-lift-check valve at each connection to the circulating-pump suction pipes, so interlocked with the main injection valves that the latter must be closed before the former can be opened, as a safeguard against flooding the main drain from the sea.

As auxiliary connections to the main drain proper, there are two 5½-inch suction branches, via the secondary drain, to the engine-room fire and bilge pumps. These branches are each fitted with a Macomb strainer and a stop valve at the main.

*Secondary Drain.*—The forward secondary drain extends in single line, from a manifold at frame No. 24 to the forward fireroom, where it joins the fireroom system; there being a Macomb strainer and stop valve at junction. It is 4½ inches in diameter, with full-size suction branch from the drain tank forward, and 3-inch branches from the double bottoms and forward trimming tanks.

In the firerooms the main is 5½ inches in diameter, with branches to double bottoms and fireroom bilge wells, 3 inches and 5½ inches in diameter, respectively. Each fireroom fire and bilge pump has a 5-inch suction connection from the main, and a discharge to the sea of same diameter.

Each engine room is provided with a complete secondary drainage system. The main is 5½ inches in diameter, with two full-size suction connections from the bilge wells. The two engine-room lines have a 5½-inch cross-connection between frames Nos. 101–102, from which are taken 4½ and 3-inch connections for drainage system aft and the port shaft alley, respectively; the starboard shaft-alley drain being taken off the 4½-inch branch leading aft. The fire and bilge pumps in each engine room have a 7-inch suction from the main, dividing into the two 5-inch connections—one to each pump. The combined overboard discharge is 7 inches in diameter.

In addition to the former, there is a shaft bilge pump located in each shaft alley, with 4½-inch independent suctions

from the after engine-room bilge and shaft alley, and overboard discharge of same size.

The after system is a single pipe 4½ inches diameter, with 3-inch branches, to the double bottoms and after trimming tanks. It is fitted with Macomb strainer and stop valve where it joins the engine-room system.

All suction connections from bilge wells in machinery spaces are fitted with Macomb strainers and stop-check valves operated at place only. The double-bottom suctions are fitted with stop-lift-check in way of machinery spaces, and stop-check valves elsewhere; the former being operated at place only, and the latter at place and from the protective deck.

Each hand pump* has a 2½-inch suction connection from the secondary drain—one in the forward fireroom, and the other in the port engine room.

*Chain-Locker Drains.*—A 3-inch drain is led from the bottom of the chain locker to the drain tank forward, frames Nos. 25–26, with stop valve at the tank. A 2-inch deck drain, from the windlass machinery space is also led into this pipe.

*Torpedo-Tube Drains.*—There is a 4½-inch drain pipe from each torpedo tube, combining into a common pipe of same diameter, fitted with stop valve and led to the drain tank forward.

*Turret and Deck Drains.*—2-inch deck drains are installed as required, each fitted with a valve on deck. Those forward unite into a common main of 3 and 4½ inches diameter, with stop valve at end, discharging into the forward fireroom bilge. Amidship they form a common 3½-inch pipe, with stop valve at end, discharging into the port engine-room bilge. Aft the combined line is 3, 4 and 4½ inches diameter, discharging through stop valve into the port engine-room bilge.

## TRIMMING TANKS.

There are four trimming tanks—two forward and two aft, of the following capacities:

* See Table I.

| Comp't. | Fresh water, tons. | Salt water, tons. | Comp't. | Fresh water, tons. | Salt water, tons. |
|---|---|---|---|---|---|
| A–1 | 91.70 | 94.45 | D–12 | 192.07 | 197.83 |
| A–2 | 117.70 | 121.23 | D–13 | 48.70 | 50.16 |

The forward tanks are flooded by a 6-inch sea chest between frames Nos. 10 and 11, starboard side, and those aft by one of same size between frames Nos. 124 and 125, starboard side. A 4-inch pipe, with globe valve adjacent to sea valve, is led to each tank. The globe valves are operated at place and from the protective deck.

The tanks are pumped out through the secondary drain, a 3-inch connection being provided for each pair of tanks, connected to tanks A–2 and D–11 only. Tanks A–1 and D–12 have no direct suction connections, being drained into the former, respectively, through 3-inch angle valves operated from the protective deck.

## FIRE MAIN.

The fire main is supplied by eight fire and bilge pumps, located in the engine and firerooms. The two distiller circulating pumps can also be used on the fire main in emergencies.

The main is entirely below the protective deck, and extends throughout the machinery spaces and forward to frame No. 42, in two 6-inch lines, port and starboard. It is cross-connected at frames Nos. 42, 99 and 104, and supplied by two 6-inch risers from the engine-room fire and bilge pumps, and four 5-inch risers from the fireroom fire and bilge pumps, with cutout valves at the main.

Forward of frame No. 42 the main extends in single line to frame No. 18, in size 6 and 4 inches diameter.

Aft of the engine rooms a single line, 6, 5 and 4 inches in diameter, leads astern to frame No. 125, where it joins the independent sanitary system for the crew's water closets and washroom aft. There is a stop valve where it joins the latter.

On the protective deck at frames Nos. 41 and 104, starboard, there are bypasses to the sanitary system. The bypasses are 6 inches in diameter and fitted with locked valves.

At frame No. 77, protective deck, amidship, there are two 4½-inch emergency connections for supplying circulating water to the distillers, which are also used for supplying the fire main by the distiller circulating pumps.

There are 6-inch flooding conections to all coal bunkers, branches to magazine flooding and sprinkling, and risers as required, leading to fire plugs on the decks above, distributed as follows:

| | | | |
|---|---|---|---|
| Lower platform | 1 | Gun deck | 25 |
| Upper platform | 14 | Main deck | 15 |
| Protective deck | 11 | Superstructure deck | 2 |
| Berth deck | 1 | | — |
| Half deck | 3 | Total | 72 |

SANITARY AND FLUSHING SYSTEM.

The flushing main is 6 inches in diameter, and is supplied by six 5-inch risers from the fire and bilge pumps, and two 4-inch bypasses, one forward and one aft, from the fire main.

It is carried in the protective-deck space, close under the gun-deck beams, and extends from frames Nos. 41 to 104, at which point it reduces in size to 4 inches and continues aft to frame No. 121, where it rises to the gun deck and leads aft to frame No. 129, joining the crew's independent flushing main, with stop valve at junction.

Branches, as required, are led to the chief petty officers' washroom and water closets, sick-bay bath, laundry, general mess pantry, galleys, bakery, firemen's washrooms, junior, warrant and wardroom officers' lavatories and water closets, etc.

Aft there is an independent system for the exclusive use of the crew's washroom and water closets. This system is supplied by two motor-driven, direct-connected, centrifugal pumps*, of about 500 gallons per minute capacity each. The main is 5 inches in diameter, with the necessary branches to the plumbing fixtures.

*See Table I.

## FRESH-WATER SYSTEM.

The following fresh-water tanks are provided as listed below:

| Compartment or tank. | Location. | Between frames. | Gallons. |
|---|---|---|---|
| A-6 | Hold-port | 18-21 | 9,684 |
| A-7 | " starbd. | 18-21 | 9,684 |
| A-8 | " port | 21-24 | 11,847 |
| A-9 | " starbd. | 21-24 | 11,847 |
| Main gravity tank | Top of chart house | 51-53 | 2,000 |
| Aux. gravity tank | Top of deck house | 60-62 | 150 |
| Firemen's supply tank | Top of deck house, port | 64-65 | 450 |
| Do. | Do. stbd. | 61-62 | 450 |
| Chief petty officers supply tank | Gun deck, starbd | 122-123 | 150 |
| Mess attendants' washroom supply tank | Gun deck, starbd | 61-62 | 100 |
| Mess attendants' lavatory supply tank | Gun deck, port | 27 | 50 |
| Battle dressing-station supply tank | Protective deck, stbd | 49-50 | 125 |
| Do. | Do. | 111-112 | 125 |
| Laundry tank | Laundry | ..... | 200 |

The hold tanks have two 2½-inch filling connections from the ship's sides forward, and also a 2-inch connection from the distiller main. From these tanks water is pumped, through the fresh-water main, to the main gravity tank, by means of two triple-plunger electric-driven pumps.* There is also a 1½-inch discharge connection from these pumps for filling the auxiliary gravity tank, which supplies water, through an independent system, to the officers' quarters above the main deck.

The main gravity tank supplies the entire fresh-water system, except as noted in preceding paragraph, including the various supply tanks. In case the gravity tank is out of order, all parts of the ship supplied by it may be supplied by the pumps direct through the main. The main is 3 inches in diameter, and extends from frames Nos. 24 to 40 on the half deck,

*See Table I.

where it descends to the protective deck and continues aft full size to frame No. 112. It has a 3-inch branch to the main gravity tank and other branches as required to the supply tanks, officers' baths, pantries, galleys, bakery, sick bay, crew's washroom, scuttle-butts, etc.

### MAGAZINE FLOODING AND SPRINKLING.

There are three flooding systems for the three groups of magazines—forward, amidship and aft. Each system floods the magazines of its group on the lower platform, while magazines on the upper platform are flooded or sprinkled from the fire main.

Each system is provided with a 9-inch sea chest, from which pipes are led, with branches to the various magazines as required. Adjacent to the sea connections are stop valves operated at place and from the protective deck.

A sprinkling system is fitted in all magazines. It is supplied by the fire main, and in the magazines consists of 2½-inch brass pipe perforated on the under side, so that each powder tank can be sprinkled. There are cutout valves for each magazine operated at place and from the protective deck.

### MAGAZINE COOLING SYSTEM.

Provision is made for supplying artificially cooled air for all magazines, except those containing saluting powder and small-arms ammunition. There are three systems supplying, respectively, the forward, middle and after groups of magazines; each system being supplied by its own ventilating blower. The blowers are electric-driven Sturtevant fans, capacity 6,000 cubic feet of air per minute each, and direct connected to 7.75-horsepower Diehl motors.

The air from the blowers is passed through cooler boxes, one for each group of magazines. The cooler boxes are fitted with pipe coils, through which the $CO_2$ gas from the refriger-

ating plant is caused to circulate, thus cooling the air delivered to the magazines as it traverses the cooling coils.

The magazines are clothed where necessary to prevent the conduction of heat.

## REFRIGERATING PLANT.

The refrigerating apparatus is of the $CO_2$ type, as manufactured by the American Carbide Machinery Co. It consists of six horizontal, electric-driven compressors, direct connected to the motors—15-horsepower units of the Diehl type, designed to run at from 80 to 110 revolutions per minute. Each compressor is capable of producing the cooling effect of 6 tons of ice per day.

The installation is divided into two independent systems—one forward, on the protective deck, frames Nos. 18 to 24 port, for the exclusive use of the forward magazines. It consists of one compressor, together with its circulating pump,* condenser and the necessary piping and fittings. The other, or main plant, is located in the refrigerating-machinery room on the protective deck, port of the engine hatches. There are five ice machines in this group, arranged so that any or all machines can be used on the refrigerating rooms, ice-making tank or magazines. There is one large condenser for this group of compressors, together with an electric-driven circulating pump,* two electric-driven brine circulating pumps,* a brine cooler and all the necessary piping and fittings. The refrigerating rooms, ice-making tank and scuttle butts are brine cooled; the brine, previously cooled in the brine cooler, being forced through the system by the brine circulating pumps and returned to the cooler, from which the cycle is repeated. The brine cooler and the after and midship magazines cooler boxes are cooled by the $CO_2$ gas, which is taken from the condenser liquid receiver by three independent systems, circulated through the coolers and returned to the compressors.

*See Table I.

There are four refrigerating rooms, on the protective deck, frames Nos. 104–113 port side. The crew's compartment serves as vestibule for the butter and meat rooms, and the officers' cooling room has no vestibule, its door opening direct to the passage. A hatch is provided through the gun deck for charging the meat room. The rooms are cork insulated in the usual manner.

### RESERVE-FEED TANKS AND FILLING CONNECTIONS.

The following double bottom compartments are fitted up as reserve-feed tanks:

| Comp't. | Fresh water, tons. | Comp't. | Fresh water, tons. |
|---|---|---|---|
| B–92 | 53.78 | B–95 | 53.69 |
| B–93 | 53.78 | B–96 | 53.78 |
| B–94 | 53.78 | B–97 | 53.69 |

A 4-inch filling pipe is led across the ship on the gun deck, frame No. 64, both outboard ends of which are carried through the main deck and fitted with two 2½-hose valves each. Amidship it has a 3-inch by-pass filling connection to the fresh-water main, fitted with locked valve, and a 4-inch branch led down into fireroom No. 3, where it joins the filling and suction manifold, through which the various compartments are filled.

### FUEL-OIL TANKS AND FILLING CONNECTIONS.

The double bottoms under the machinery spaces, frames Nos. 78 to 99, are arranged for carrying fuel oil as follows:

| Comp't. | Gallons. | Comp't. | Gallons. |
|---|---|---|---|
| C–90 | 21,722 | C–94 | 17,638 |
| C–91 | 21,692 | C–95 | 17,638 |
| C–92 | 18,102 | C–96 | 15,760 |
| C–93 | 18,102 | C–97 | 15,760 |

For filling the fuel-oil compartments, two 6-inch pipes are led across the ship under the main deck, with hose valves at

the ship's side, frames Nos. 99–100 starboard and 103–104 port; and on the main deck at frames Nos. 85 starboard and 88 port. These pipes are combined into an 8-inch pipe, which leads down the starboard engine hatch to the manifolds in the engine rooms, individual oil compartments being filled through the manifolds and suction pipes from the tanks.

An 8-inch overflow pipe is fitted to the filling pipe in engine hatch, discharging overboard at frame No. 89, gun deck, port side.

COAL–BUNKER CAPACITIES.

| Comp't. | Tons. | Comp't. | Tons. |
|---|---|---|---|
| B–5–P | 90.6 | B–102 | 81.7 |
| B–5–S | 91.5 | B–103 | 81.9 |
| B–7–P | 145.8 | B–104 | 84.4 |
| B–7–S | 145.8 | B–105 | 84.4 |
| B–9–P | 163.3 | B–106 | 80.5 |
| B–9–S | 163.3 | B–107 | 80.6 |
| B–11–P | 151.0 | B–108 | 78.9 |
| B–11–S | 151.0 | B–109 | 78.9 |
| B–13–P | 135.1 | B–110 | 97.3 |
| B–13–S | 135.1 | B–111 | 97.3 |
| B–15–P | 134.3 | B–112 | 96.6 |
| B–15–S | 134.3 | B–113 | 96.6 |
| B–17–P | 105.7 | | |
| B–17–S | 105.7 | Total | 2,892.5 |

VENTILATING SYSTEM.

Artificial ventilation is provided where necessary for all quarters, living spaces, passages, storerooms, magazines, engine rooms, dynamo rooms, evaporator room, etc. There are forty-six ventilating fans and motors, each on its own circuit. In general air is supplied on the plenum system to the different compartments requiring ventilation. The toilet spaces are also provided with the exhaust system. The ventilating fans were made by the B. F. Sturtevant Co., Boston, Mass., and

the motors by the Diehl Mfg. Co., of Elizabethport, N. J. The particulars of ventilating fans and motors are given below:

| No. | H.P. of each motor. | Capacity of each fan, cubic feet per minute. | R.P.M. | Remarks. |
|---|---|---|---|---|
| 4 | 0.88 | 1,000 | 1,560 | Inclosed motor. |
| 5 | 2.19 | 2,500 | 1,055 | Open motor. |
| 2 | 2.19 | 2,500 | 1,055 | Inclosed motor. |
| 3 | 3.50 | 4,000 | 835 | Open motor. |
| 2 | 3.50 | 4,000 | 835 | Open motor. Searchlight rheostat-room ventilation. |
| 3 | 4.37 | 5,000 | 745 | Open motor. |
| 10 | 4.80 | 5,000 | 790 | Inclosed motor. Turret ventilation. |
| 3 | 7.75 | 6,000 | 820 | Open motor. Magazine cooling systems. |
| 7 | 7.00 | 8,000 | 590 | Open motor. |
| 3 | 8.75 | 10,000 | 530 | Open motor. |
| 4 | 10.50 | 12,000 | 485 | Inclosed motor. Engine-room ventilation system. |

Heater boxes are fitted in the ventilating ducts to all quarters below the main deck, crew's space, etc., for heating the incoming air in cold weather; no other heating apparatus being provided for these spaces.

## HEATING SYSTEM.

All staterooms and quarters below the main deck, crew's space, etc., are heated by the thermo-tank ventilating system; heater boxes provided with suitable steam coils being placed in the air ducts for heating the air supplied to these compartments. The remaining portions of the vessel, *i.e.*, quarters on the main, superstructure and bridge decks, are heated by the customary pipe-coil steam radiators.

The heating system is divided into two main sections, one forward and one aft. The forward section is taken off the main steam cross-connection pipe in fireroom No. 2, starboard side, with stop and reducing valves adjacent to the steam line, and is subdivided into five independent circuits as follows:

| Circuit No. | Description. | Working pressure. Pounds. |
|---|---|---|
| 1 | Crew's and officers' galleys and bakery..... | 20 |
| 2 | Bath, shower and lavatory-water heaters forward of frame No. 74; pantries and fresh-water tanks ..... | 20 |
| 3 | Heater boxes (2) for quarters forward of forward diagonal armor on half and gun decks ..... | 50 |
| 4 | Heater boxes (4) for space between frames Nos. 42–47, upper platform; frames Nos. 42–50, protective deck, and between forward and after diagonal armor on gun deck | 50 |
| 5 | Radiators in quarters on main, superstructure and bridge decks, and fresh-water tanks on chart house ..... | 50 |

The after section is taken off the auxiliary steam loop in engine rooms, with stop and reducing valves at the line, and is subdivided into eight independent circuits as given below:

| Circuit No. | Description. | Working pressure. Pounds. |
|---|---|---|
| 6 | Heater boxes (2) for space between after diagonal armor and frame No. 104, gun deck | 50 |
| 7 | Heater boxes (2) for space between frames Nos. 76–104, protective deck..... | 50 |
| 8 | Heater box for steering room, steering-gear room and pump rooms..... | 50 |
| 9 | Heater boxes (3) for quarters and space aft of frame No. 104, protective deck..... | 50 |
| 10 | Heater boxes (2) for crew's space between frames Nos. 113–130, gun deck..... | 50 |
| 11 | Heater box for hospital space..... | 50 |
| 12 | Laundry and general mess pantry..... | 20 |
| 13 | Bath, shower and lavatory heaters aft of frame No. 98..... | 20 |

The drains, from circuits Nos. 3, 4 and 5, are collected into a manifold and led to a common trap in fireroom No. 2. Circuit No. 1 has an independent trap located in the same fireroom. The drains from bath heaters, etc., circuits, Nos. 2 and 13, are led direct to the auxiliary exhaust line. Circuits Nos. 6, 7, 8, 9, 10 and 11 are led to a drain manifold in the port engine room, which connects to a trap in the same engine room. There is also an independent trap in the port engine room for the drain from circuit No. 12. All traps discharge into the low-pressure trap discharge main.

The following is a list of the heater boxes in the ventilating ducts:

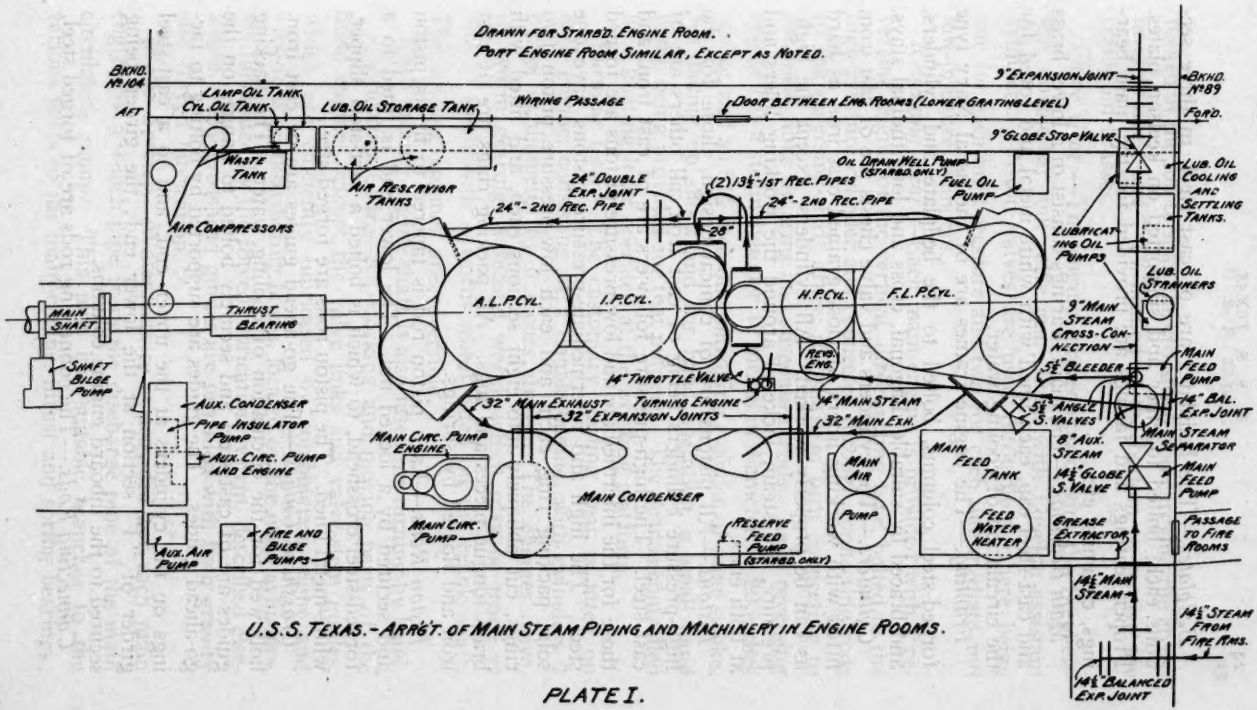
| No. | Location. | Between frames. | Heating Surf. Sq. ft. each. | Steam from |
|---|---|---|---|---|
| 2 | Gun Deck | 92–93–P&S | 187.3 | Circuit No. 6. |
| 1 | Do. | 111–112–S | 56.7 | Circuit No. 11. |
| 2 | Do. | 112–113–P&S | 130.1 | Circuit No. 10. |
| 1 | Do. | 124–125–A | 174.1 | Circuit No. 9. |
| 1 | Prot. Deck | 32–33–P | 355.5 | Circuit No. 3. |
| 1 | Do. | 34–35–S | 355.5 | Do. |
| 1 | Do. | 45–46–S | 131.2 | Circuit No. 4. |
| 1 | Do. | 46–47–P | 161.3 | Do. |
| 2 | Do. | 51–52–P&S | 259.4 | Do. |
| 2 | Do. | 91–92–P&S | 115.8 | Circuit No. 7. |
| 1 | Do. | 112–113–P | 69.4 | Circuit No. 9. |
| 1 | Do. | 112–113–S | 82.8 | Do. |
| 1 | Steering room | 124–126–P | 83.2 | Circuit No. 8. |

(NOTE.—Letters P, S and A, after frame numbers, indicate port, starboard and amidship, respectively).

## MAIN ENGINES.

There are two main engines designed to develop collectively 28,100 indicated horsepower, when making 125 revolutions per minute. They are placed abreast in two separate watertight compartments, as shown in Plate I.

The engines are of the vertical, inverted-cylinder, direct-acting, four-cylinder, triple-expansion type, turning outboard when going ahead. The order of the cylinders, beginning forward, is forward low-pressure, high-pressure, intermediate-pressure and after low-pressure. All crank angles are 90 degrees, the sequence of cranks being high-pressure, intermediate-pressure, forward low-pressure and after low-pressure.




U.S.S. TEXAS. - ARR'G'T. OF MAIN STEAM PIPING AND MACHINERY IN ENGINE ROOMS.

PLATE I.

*Bedplates.*—The bedplates are of cast steel, in three sections each, bolted together and supported on keelson plates. Proper seatings and facings are provided for the main bearings, columns, etc.

*Main Bearings.*—The main bearings consist of a lower brass and cast-steel cap, each lined with white metal and cored for the circulation of cooling water.

*Framing.*—The engine frames are of the usual Navy type forged-steel columns, bolted to the bedplate and cylinders, and braced by suitable diagonal, cross and longitudinal stays.

*Cylinders.*—The cylinders and valve chests are of cast iron, fitted with working liners of close-grained cast iron as hard as can be properly worked. All cylinders, except the high-pressure, are steam jacketed around the working liners and at both ends.

*Pistons.*—All pistons are of conical design, those for the high-pressure cylinders being of cast iron, all others are of cast steel. The high-pressure followers are of cast iron, and those for the intermediate and low-pressure pistons are forged steel. The high and intermediate-pressure pistons have one solid packing ring each, and each low-pressure piston two rings, cut obliquely into eight sections each, and fitted with brass tongue pieces and lugs. All packing rings are of cast iron and floated by springs.

*Piston Rods.*—Each piston rod is tapered to fit its piston and secured by a locked nut. The lower end is fitted to a forged-steel crosshead, to which is bolted a cast-steel slipper, white-metal lined. The piston rods are forged steel.

*Crosshead Guides.*—The go-ahead guides are of cast iron, hollowed for the circulation of cooling water. The backing guides are of cast steel and securely bolted to flanges on the go-ahead guides. The guides are supported by bolting to facings on the cylinders at the upper end, and to a cast-steel girder of "I" section at the lower end; the girder being secured to the inboard engine columns.

*Connecting Rods.*—The connecting rods are of forged steel,

forked at the top to span the crosshead and carrying the crosshead brasses, and "T" headed at the bottom to receive the crank-pin brasses.

*Valve Gear.*—The engines are equipped with the Stephenson, double-bar link valve gear, fitted with Lovekin assistant cylinders. Piston valves are used throughout; there being one for each high-pressure and two for each intermediate and low-pressure cylinders.

*Reversing Gear.*—Each main engine is provided with a reversing engine of the vertical, direct-acting type, bolted to the high-pressure cylinder and connected through connecting rods to the reversing-shaft arms; the shaft in turn connecting to the main links by arms and suspension rods. Each reversing engine has a steam cylinder 17 inches in diameter by 22 inches stroke, and an 8½-inch oil-controlling cylinder of same stroke, for taking up shock and for hand operation, a small hand pump being provided for that purpose. The gear is controlled by a floating lever operated at the working platform.

*Turning Gear.*—The customary turning gear is fitted on each engine, consisting of a double engine, with cylinders 7 inches in diameter by 5 inches stroke. The engines drive, by worm gearing, a second worm, which may be made at will to mesh with a worm wheel fitted on the crank shaft. The turning engines are reversible.

Each turning-engine shaft is also fitted for turning by hand.

*Lubricating Gear.*—All working and moving parts of the main engines, except the valve links and valve-stem guides, which are efficiently lubricated by combination sight and wick-feed oil-distributing boxes, located on the main cylinders, are lubricated by the forced-lubrication system, described elsewhere. The crosshead guides are provided with both gravity and forced lubrication.

*Water Service.*—Water service is provided for each main engine by a 4-inch pipe from the discharge pipe of the main circulating pump. This pipe has suitable branches to the various parts of the main engine, thrust and spring bearings;

the discharge being returned to the suction side of the same pump.

*Working Platform.*—Owing to the fact that the forced-lubrication system requires the crank pits to be totally inclosed by a thin galvanized sheet-steel casing extending up nearly to the bottoms of the cylinders, the working platforms are on the first grating above the engine-room floor, there being a door through the center-line bulkhead, at this level, between the engine rooms.

### *Main Engine Data.*

| | |
|---|---|
| Working pressure at H.P. chest, pounds gage | 265 |
| Revolutions per minute, designed | 125 |
| Indicated horsepower, total designed | 28,100 |
| Diameter of H.P. cylinder, inches | 39 |
| I.P. cylinder, inches | 63 |
| L.P. cylinders (2), inches | 83 |
| Stroke, inches | 48 |
| Ratio, I.P. to H.P. | 2.61 |
| L.P. to H.P. | 9.06 |
| L.P. to I.P. | 3.47 |
| Total expansions | 10.76 |

| | Per cent. of volume. | | Linear, inches. | |
|---|---|---|---|---|
| Cylinder clearances: | Top. | Bottom. | Top. | Bottom. |
| Starboard, H.P. | 13.17 | 15.23 | $\frac{3}{8}$ | $\frac{17}{32}$ |
| I.P. | 12.14 | 15.80 | $1\frac{3}{32}$ | $1\frac{5}{32}$ |
| F.L.P. | 13.14 | 17.07 | $2\frac{11}{32}$ | $2\frac{21}{32}$ |
| A.L.P. | 13.31 | 16.59 | $\frac{9}{16}$ | $\frac{9}{16}$ |
| Port, H.P. | 13.14 | 15.19 | $\frac{9}{16}$ | $\frac{5}{8}$ |
| I.P. | 12.13 | 15.79 | $1\frac{3}{32}$ | $1\frac{1}{32}$ |
| F.L.P. | 13.14 | 17.07 | $\frac{3}{8}$ | $\frac{31}{32}$ |
| A.L.P. | 13.31 | 16.59 | $\frac{27}{64}$ | $\frac{15}{32}$ |

| Valves and valve settings: | H.P. | | I.P. | | L.P. (each). | |
|---|---|---|---|---|---|---|
| Number and type of valves | 1 piston. | | 2 piston. | | 1 piston. | |
| Diameter of valves, inches | $22\frac{1}{16}$ top. 22 bottom. | | $24\frac{1}{16}$ top. 24 bottom. | | $34\frac{1}{16}$ top. 34 bottom. | |
| Travel of valves, inches | 10 | | 11 | | 12 | |
| Inside or outside steam | Inside. | | Outside. | | Outside. | |
| | Top. | Bot. | Top. | Bot. | Top. | Bot. |
| Width of port, inches | $3\frac{1}{4}$ | $3\frac{1}{4}$ | $3\frac{3}{4}$ | $3\frac{3}{4}$ | $4\frac{1}{4}$ | $4\frac{1}{4}$ |
| Steam opening, linear, inches | $3\frac{5}{16}$ | $3\frac{5}{16}$ | $3\frac{5}{16}$ | $3\frac{7}{16}$ | $2\frac{13}{16}$ | 3 |
| Exhaust opening, linear, inches | Full port. | | Full port. | | Full port. | |
| Steam lap, inches | $1\frac{13}{16}$ | $1\frac{1}{16}$ | $2\frac{5}{16}$ | $2\frac{1}{16}$ | $3\frac{1}{16}$ | 3 |
| Exhaust lap, inches | $-\frac{7}{16}$ | $\frac{1}{4}$ | $-\frac{9}{16}$ | $\frac{1}{4}$ | $-\frac{1}{8}$ | $\frac{1}{4}$ |
| Steam lead, linear, inches | $\frac{1}{16}$ | $\frac{13}{16}$ | $\frac{7}{8}$ | 1 | $1\frac{1}{16}$ | $1\frac{1}{4}$ |
| Cutoff, decimal of stroke | 85.15 | 78.65 | 81.5 | 73.7 | 67.06 | 56.9 |
| mean | 81.9 | | 77.6 | | 61.98 | |

| | |
|---|---|
| Piston rods, diameter, outside, inches | 08¼ |
| axial hole, inches | 1½ H.P. and I.P., 4 L.P. |
| Connecting rods, diameter, crosshead end, inches | 07½ |
| crank end, inches | 09¼ |
| axial hole, inches | 1½ H.P. and I.P., 4 L.P. |
| length between centers, inches | 96 |
| crank ratio | 4 |
| Diameter of throttle valve, inches | 14 |
| 1st receiver pipes (2), inches | 13½ |
| 3d receiver pipe, inches | 24 |
| main exhaust pipes (2), inches | 32 |
| live-steam pipes to receivers, inches | 03 |

## SHAFTING AND BEARINGS.

There are two lines of shafting, each consisting of a crank shaft in two sections, a thrust shaft, one piece of line shafting, a stern-tube shaft and a propeller shaft, all supported by suitable bearings.

*Crank Shafts.*—The crank shafts are solid-forged, hollow shafts, in two sections each; bolted together by disc couplings, and carried in seven main bearings. The forward section includes the H.P. and F.L.P. cranks, which are opposite, as are also the I.P. and A.L.P., on the after section. All crank angles are ninety degrees, the two sections being connected with the H.P. crank leading followed by the I.P. crank.

*Thrust Shafts.*—The thrust shafts are fitted to the usual thrust bearings of the adjustable horse-shoe type, with steady bearings at each end for supporting the shafts. The shafts are hollow with thrust collars and coupling discs forged integral with the shafts.

*Line Shafts.*—These shafts are hollow forged, with disc couplings at either end, and each is carried by one steady bearing.

*Stern-Tube Shafts.*—The stern-tube shafts are covered with a composition casing within the tern tubes and at bearings. They are hollow forged, secured to the line shafts at inboard end by special disc couplings, and to the propeller shafts at outboard end by the usual sleeve couplings, the shafts being taper turned to suit. Each stern tube is provided with two lignum

vitae lined bearings for supporting the shafts. There is also one spring bearing in each shaft alley supporting the overhanging ends of these shafts.

*Propeller Shafts.*—Each propeller shaft is carried by two lignum vitae lined bearings, fitted to the forward and after struts. The shafts are hollow forged, taper turned at both ends to suit the propeller hub and sleeve coupling, and composition bushed at the bearings.

*Inboard Coupling.*—The inboard coupling consists of a sleeve secured to the stern-tube shaft by four keys. Back of the sleeve is a collar made in halves and secured to the sleeve and to the coupling disc on the line shaft by fitted bolts.

*Outboard Coupling.*—The outboard coupling is of the solid-sleeve type, taper bored to fit the shafts, and secured to each shaft by two feather keys and one cross key.

*Shaft Data.*

| | |
|---|---|
| Crank shafts, length, forward section, feet and inches | 17–03¼ |
| after section, feet and inches | 19–04 |
| diameter, inches | 18¾ |
| axial hole, forward section, inches | 11½ |
| after section, inches | 10½ |
| number of cranks | 4 |
| throw of cranks inches | 24 |
| crank angles, degrees | 90 |
| Crank pins, length, inches | 22 |
| diameter, inches | 19½ |
| axial hole, forward section, inches | 12 |
| after section, inches | 10 |
| Thrust shafts, length, feet and inches | 16–04¼ |
| diameter, inches | 17¾ |
| at bearings, inches | 18 |
| axial hole, inches | 11½ |
| collars, number | 14 |
| thickness, inches | 02 |
| space between, inches | 04 |
| outside diameter, inches | 27½ |
| inside diameter, inches | 18 |
| bearing surface, square inches | 4,501.14 |
| Line shafts, length, feet and inches | 18–00 |
| diameter, inches | 18 |
| axial hole, inches | 11½ |

| | |
|---|---|
| Stern-tube shafts, length, feet and inches | 51–08½ |
| diameter, inches | 18½ |
| axial hole, inches | 11 |
| Propeller shafts, length, feet and inches | 55–09¾ |
| diameter, inches | 18½ |
| axial hole, inches | 11 |
| Coupling discs, diameter, inches | 33 |
| thickness, inches | 04 |
| Inboard coupling, diameter of sleeve outside, inches | 33 |
| inside, inches | 20½ |
| length of sleeve, inches | 12 |
| thickness of collars, inches | 4 |
| Coupling bolts, number each coupling | 8 |
| diameter (taper)* at face of coupling, inches | 03⅞ |
| Outboard couplings, length of sleeve, inches | 73⅞ |
| diameter of sleeve, inches | 24¼ |

## *Bearing Data*

| | |
|---|---|
| Main bearings (white-metal lined): | |
| Number each engine | 7 |
| Diameter, inches | 18¾ |
| Length, inches | two of 15, three of 24 and two of 27¾ |
| Thrust bearings (white-metal lined bearings and shoes): | |
| Steady bearings, number each | 2 |
| diameter, inches | 18 |
| length, inches | 19 |
| Thrust shoes, number each | 14 |
| effective surface, square inches | 2,986.1 |
| Line-shaft bearings (white-metal lined): | |
| Number each engine | 2 |
| Diameter, inches | 18 |
| Length, inches | 26 |
| Stern-tube bearings (lignum vitae lined): | |
| Number each engine | 2 |
| Forward bearing, diameter, inches | $20\frac{2}{16}$ |
| length, inches | 48 |
| After bearings, diameter, inches | 20⅝ |
| length, inches | 65 |
| Strut bearings (lignum vitae lined): | |
| Number each engine | 2 |
| Forward bearing, diameter, inches | $20\frac{11}{16}$ |
| length, inches | 38 |
| After bearing, diameter, inches | 20¾ |
| length, inches | 63 |

*Parallel bolts for inboard coupling.

## PROPELLERS.

There are two twin-screw, three-bladed, propellers of the adjustable-pitch, detached-blade type. They are of manganese-bronze and the blades are secured to the hubs by seven 4¾ tap bolts each. The blades are machined true to pitch and the hubs have a taper fit on the shafts, and are secured by a key and nut.

### *Propeller Data.*

| | |
|---|---|
| Diameter of propeller, feet and inches | 18–07¾ |
| hub, feet and inches | 4–06 |
| Pitch as set, feet and inches | 19–11⅞ |
| adjustable from, feet and inches | 19–00¾ to 20–05½ |
| Ratio of diameter to pitch | 0.9342 |
| Area, projected, square feet | 87.9 |
| helicoidal, square feet | 99.18 |
| disc, square feet | 271.966 |
| Ratio, projected to disc area | 0.323 |
| helicoidal to disc area | 0.365 |
| Heighth of lower tip of blade above keel, inches | 8⅛ |
| Immersion of upper tip of blade, inches | 110⅛ |

## MAIN CONDENSING APPARATUS.

*Main Condensers.*—There is one main condenser of cylindrical form for each main engine, with tubes rolled into the tube sheets at one end and gland packed at the other. The principal dimensions follow:

| | |
|---|---|
| Inside diameter, feet and inches | 7–09 |
| Thickness of shell (steel), inch | $00\frac{7}{16}$ |
| Length between tube sheets, feet and inches | 15–00 |
| Thickness of tube sheets, inches | 01⅛ and 1⅛ |
| Tubes, number | 5,340 |
| diameter, outside, inch | 00⅝ |
| thickness, inch | 00.065 |
| Cooling surface, square feet | 13,104 |
| Diameter of main exhaust nozzles (2), inches | 32 |
| auxiliary exhaust nozzle, inches | 08 |
| air-pump suction, inches | 12 |
| circulating-water inlet and outlet, inches | 23 |

**Main Air Pumps.*—Each main condenser is provided with a Blake, vertical, twin, bucket, single-acting air pump, with steam and water cylinders 14 and 35 inches diameter, respectively, by a common stroke of 21 inches. The suction nozzle is 12 inches and the discharge nozzle 10 inches in diameter.

**Main Circulating Pumps and Engines.*—There is one centrifugal circulating pump for each main condenser, driven by a vertical compound engine. The engine is provided with a self-contained forced-lubrication system, with branch connection from the main engine system. The principal dimensions of the pump and engine are as follows:

| | |
|---|---|
| Capacity of pump, gallons per minute | 21,000 |
| Diameter of suction nozzle (2), inches | 17 |
| discharge nozzle, inches | 23 |
| impeller, inches | 42 |
| H.P. cylinder, inches | 19 |
| L.P. cylinder, inches | 20 |
| Stroke, inches | 10 |
| Revolutions per minute | 235 |

*Feed and Filter Tank.*—A feed and filter tank of 3,920 gallons capacity is located in each engine room, outboard side and forward of the main condenser. The filter chamber is in the top of the tank and has a capacity of about 814 gallons. The filter has an inner bottom of loose perforated plates and is divided into compartments, in which is placed the filtering material, by vertical division plates. These partitions are so arranged that the water in passing through the filter will flow under and over in succession, thus assuring the filtering material being always submerged.

Each tank is provided with the following connections:

1 10-inch main air-pump discharge;
1 8-inch cross-connection;
2 5½-inch main feed-pump suctions;
1 4-inch auxiliary air-pump discharge;

*See Table I.

# TABLE I.

## PUMPS AND PUMP CONNECTIONS.

| No. | Pumps | Size (Ins.) | Type | Suction Pipes From— | | Discharge Pipes To— | | Location |
|---|---|---|---|---|---|---|---|---|
| | | | | Ins. | | Ins. | | |
| 2 | Main Air | (2) 14 x (2) 35 x 21 | Blake, Twin, Vertical, Single-Acting | 12 | Condenser | 10 | Feed Tank | 1 in each Eng. Room |
| 2 | Main Circulating | 42 Impeller; 10 x 20 x 42 Eng. | Horizontal, Centrifugal; Vert. Compound Eng. | (2) 17<br>15½ | Sea<br>Main Drain | 23<br>4 | Condenser<br>Water Service | Do. |
| 4 | Main Feed | 15½ x 9½ x 24 | Blake, Vert., Double-Acting, Single | 5½<br>5½ | Feed Tanks<br>Channel Way Cross Connection | 5 | Main Feed Line | 2 in each Eng. Room |
| 4 | Aux. Feed | Do | Do | 5½<br>4<br>4<br>2½ | Fr. & D. Suction Pipe Ship's Sides (a)<br>Reserve Fd. Tanks<br>Hose Connection | 5<br>3½<br>2½ | Main Feed Line<br>Reserve Fd. Tanks (b)<br>Each Boiler in Compt. with Pump | 1 in each Boiler Room |
| 4 | Fire and Bilge | 12 x 10 x 10 | Do. | 5<br>5<br>1½<br>4½ | Sea<br>Drainage<br>Hose Connection<br>Fuel Oil Tanks (c) | 5<br>5<br>5<br>2½ | Fire Main<br>Overboard<br>Sanitary Main (b)<br>Hose Connection | 2 in each Eng. Room |
| 4 | Fire, Bilge and Ash Ejector | 14 x 8 x 12 | Blake, Vert., Double-Acting, Duplex | 5<br>5<br>1½<br>2½ | Sea<br>Drainage<br>Hose Connection<br>Do. | 5<br>5<br>5<br>4<br>2½ | Fire Main<br>Overboard<br>Sanitary Main<br>Ash Ejectors<br>Hose Connection | 1 in each Boiler Room |
| 4 | Forced Lub. Service | 7 x 8 x 8 | Blake, Vert., Double-Acting, Single | 3½<br>3½ | Oil Tanks<br>Oil Well (d) | 3<br>2½ | Settling Tanks (d)<br>Main Bearings | 2 in each Eng. Room |
| 2 | Forced Lub. Return | Do | Do | 4 | Oil Well | 3½ | Settling Tanks | 1 in each Eng. Room |
| 1 | Oil Well Drain | 2 x 1½ x 2½ | Blake, Vert., Duplex, D-Acts. | 1 | Oil Well | ½ | Bilge | Starb'd. Eng. Room |
| 2 | Fuel Oil Supply | 14 x 9 x 12 | Blake, Vert., Double-Acting, Single | 5<br>5 | Storage Tanks<br>Ship's Sides | 4½<br>4½<br>4½ | Burner Supply<br>Storage Tanks<br>Overboard | 1 in each Eng. Room |
| 2 | Aux. Air | 7½ x 14 x 12 | Do. | 6 | Aux. Condenser | 4 | Main Feed Tanks | Do. |
| 2 | Aux. Circulating | 28 Impeller; 5 x 4 Engine | Hor., Centrifugal Vert., Single Eng. | 5 | Sea | 5 | Aux. Condenser | Do. |
| 2 | Dynamo Cond. Air | 9 x (2) 18 x 12 | Blake, Twin, Vert., Single-Acting | 7 | Dynamo Cond. | 6 | Dynamo Room Hot Well Tank | 1 in each Dyn. Cond. Room |
| 2 | Dynamo Cond. Circ. | 26 Impeller; 6 x 5 Engine | Hor. Centrifugal Vert., Single Eng. | (2) 5½ | Sea | 8 | Dynamo Cond. | Do. |
| 2 | Dyn. Rm. Hot Well | 4½ x 5 x 6 | Blake, Vert., Double-Acting, Single | 2½ | Dyn. Room Hot Well Tank | 2 | Aux. Feed Suction Main | Do |
| 2 | Evaporator Feed | Do | Do. | 2½<br>2½ | Sea<br>Dist. Circ. Disch. | 2 | Evaporators | Evap. Room |
| 2 | Distiller Circ. | 8 x 12 x 12 | Do | 6½ | Sea | 6<br>4½ | Distillers<br>Fire Main | Do. |
| 2 | Distiller Fresh Water | 4½ x 5 x 6 | Do | 2½ | Distillers | 2<br>1½<br>1½ | Fresh Water Tanks<br>Aux. Fd. Suction<br>Reserve Fd. Tanks | Do. |
| 2 | Pipe Insulator Circ. | 7 x 8 x 12 | Do | 4½ | Sea | 4 | Pipe Insulators | 1 in each Eng. Room |
| 1 | Reserve Feed | 4½ x 5 x 6 | Do | 3 | Reserve Fd Tanks | 2½<br>2 | Main Feed Tanks<br>Air Comp. Circ. Mat. | Starb'd. Eng. Rm. |
| 2 | Shaft Bilge | 9 x 6 | Plunger, Single-Acting | 4½<br>4½ | Eng. Room Bilge<br>Shaft Alley | 4½ | Overboard | 1 in each Shaft Alley |
| 2 | Brine Circ. | 2 | Jeanesville, Hor., Volute, Motor Driven | 3<br>2 | Ice Tank<br>Brine Cooler | 2 | Brine Circ. | After Ice Mach. Room |
| 1 | CO2 Cond. Circ. | 3 | Do | 4 | Sea | 3 | CO2 Condenser | Do. |
| 1 | Do. | 2 | Do. | 3 | Sea | 2 | Do. | Ford. Ice Mach. Rm. |
| 2 | Flushing | 5 | Centrifugal, Motor Driven | 6 | Sea | 5 | Crew's Independent Sanitary System | 1 each Compts. D-26-F.S |
| 2 | Fresh Water | (3) 4 x 6 | Triple Plunger, Motor Driven | 2½ | Fresh Water Tanks | 2½ | Fresh Water System | Compt. A-49-S |
| 2 | Hand | 6 x 6 | Hand, Horizontal | 2½<br>2½ | Sea<br>Drainage | 2½ | Hose Connection | 1 on U.P. Frs. 46-47 F; 1 on F.D. Fr. 103 C. |

(a) No. 3 Fire Rm. only. (b) One Pump in each Eng. Rm. (c) Portable Connection. (d) One Pump in each Eng. Rm.

1 2½-inch reserve-feed pump discharge (starbd. only);
1 8-inch overflow;
3- and 2-inch vapor pipes (combined into one 3-inch pipe).

ENGINE-ROOM AUXILIARIES.

*Auxiliary Condensers.*—In each engine room there is an auxiliary condenser connected through the auxiliary exhaust pipe to all the auxiliary machinery. The tubes are rolled into the tube sheets at one end and packed at the other.

They are of the following principal dimensions:

| | |
|---|---|
| Inside diameter, feet and inches | 2–00 |
| Thickness of shell (steel) inch | 00$\frac{5}{16}$ |
| Length between tube sheets, feet and inches | 6–06¼ |
| Thickness of tube sheets, inch | 01 |
| Tubes, number | 333 |
| diameter, outside, inch | 00⅝ |
| thickness, inch | 00.065 |
| Cooling surface, square feet | 355.24 |
| Diameter of auxiliary exhaust nozzle, inches | 06 |
| air-pump suction, inches | 06 |
| circulating-water inlet and outlet, inches | 05 |

**Auxiliary Air Pumps.*—A Blake, vertical, double acting, single, featherweight, air pump is provided for each auxiliary condenser, with steam and water cylinders of 7½ and 14 inches diameter, respectively, by a common stroke of 12 inches. The suction nozzle is 6 and the discharge nozzle 4 inches in diameter.

**Auxiliary Circulating Pumps.*—Each auxiliary condenser is equipped with a centrifugal circulating pump, driven by a vertical, single engine, fitted with a self-contained system of forced lubrication. The general dimensions of the pump and engine follow:

| | |
|---|---|
| Diameter of suction nozzle, inches | 05 |
| discharge nozzle, inches | 05 |
| impeller, inches | 28 |
| steam cylinder, inches | 05 |
| Stroke, inches | 04 |
| Revolutions per minute | 400 |

*See Table I.

*Feed-Water Heater.*—Two Reilly multicoil feed-water heaters are installed—one in each engine room. They have 319.8 square feet of heating surface each, and are connected to the main feed lines only. The heating agent is the exhaust steam, a back pressure being kept in the auxiliary exhaust line for this purpose by means of a spring-relief valve at each connection to the main and auxiliary condensers, opening toward the condenser.

**Main Feed Pumps.*—Two Blake, vertical, double-acting, single, main feed pumps are installed on the forward bulkhead of each engine room. The pumps have independent suctions from the main feed tanks in same engine room and discharge to the boilers through the feed-water heaters or bypassing same.

**Reserve Feed Pump.*—A small Blake, vertical, double-acting, single, reserve feed pump is fitted in the starboard engine room, for use in port to pump makeup feed water from the reserve feed tanks into the main feed tanks.

**Main Fire and Bilge Pumps.*—Located aft, on the outboard bulkhead, in each engine room, are two Blake, vertical, double-acting, single, fire and bilge pumps. They are arranged to draw water from the drainage system and the sea, and discharge to the fire main, sanitary system and overboard.

**Pipe Insulator Circulating Pumps.*—In each engine room there is a Blake, vertical, double-acting, single, pump, for circulating sea water around the main steam-pipe flanges at bulkheads near magazines, to prevent the transmission of heat through the ship's structure to the magazines.

## FORCED-LUBRICATION SYSTEM.

All working and moving parts of the main engines, except the valve links and valve-stem guides, are fitted with forced lubrication.

The installation in each engine room comprises three

*See Table I.

pumps,* two 500-gallon oil-settling and cooling tanks, together with the necessary piping and fittings. The settling and cooling tanks are provided with steam coils and cooling coils, the cooling agent being sea water, supplied from the pipe insulator circulating system. Each engine-room system is complete and independent, a cross connection being provided, however, for emergencies. The crank pits are of oiltight construction, fitted with an oil-drain well at the forward end, and so designed that no bilge water or dirt can enter therein. An oil trough is thus formed at the base of each engine to catch all the oil. A small pump* is provided in the starboard engine room, with suction connections to the bottom of each crankpit oil well, for pumping out any water that may have collected from the oil or through leakage. This pump discharges to the bilge. The entire engine is incased with a light galvanized sheet-steel casing, to prevent splashing and waste of oil, which is carried up to within about eighteen inches of the bottoms of the cylinders.

The plant operates as follows: One pump draws the oil from the settling and cooling tanks, and discharges same through a pipe, fitted in duplicate for emergencies, having branches to the main bearings through holes in the caps. An annular groove in the center of each main bearing provides for the proper distribution of the oil, part of which lubricates the bearings, the balance passing through a radial hole in each journal, in wake of the groove, to the crankshaft axial holes, the openings at ends of each axial hole being closed by oiltight cover plates. From the axial holes, the oil is forced through radial holes to the eccentric straps and crank-pin axial holes, and through radial holes to the surface of the crank pins. The crank-pin bearings are similarly grooved to the main bearings, all oil not used for crank-pin lubrication, passing up through brass tubes secured to the connecting rods, to the crosshead bearings. After performing its

*See Table I.

function the oil escapes at the ends of the bearings and drains to the crank pit, where it collects in the drain well and is pumped by a second pump, through oil filters, to the main supply and settling tanks, thus completing its cycle, which is indefinitely repeated as described. The third pump is a stand-by for emergencies and connected to perform the duties of either of the other two pumps.

Pressure gages and thermometers are provided at each main bearing. The system is generally operated at about 50 pounds pressure.

Branches are taken off the system for the thrust bearings and main circulating-pump engine.

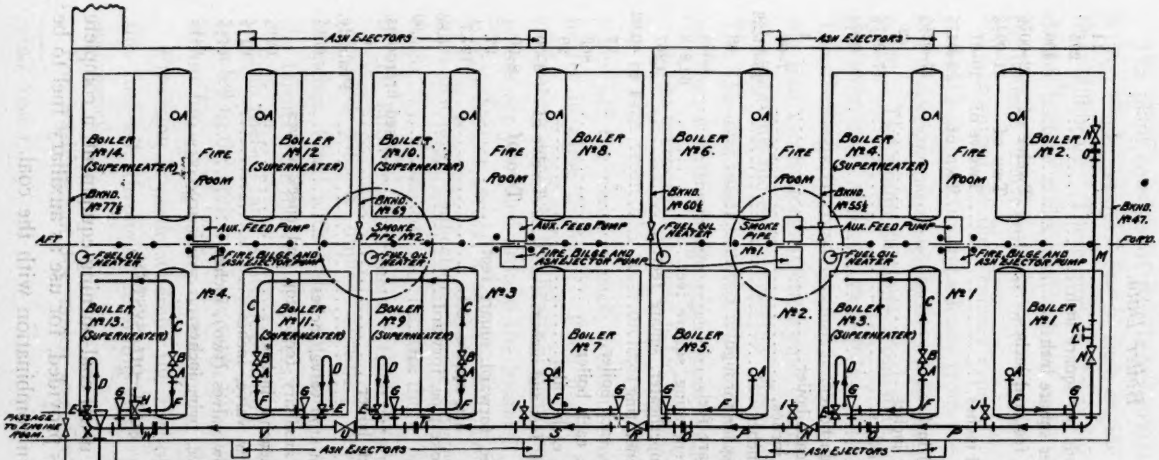
Large storage tanks of a total capacity of 2,000 gallons are installed in the engine rooms for making up leakage and other waste, and for replenishing the system when desired.

## BOILERS.

There are fourteen Babcock & Wilcox water-tube boilers, arranged in four separate watertight compartments, as shown in Plate II. There are four boilers each in firerooms Nos. 1, 3 and 4, and two boilers in fireroom No. 2. Eight of the boilers, Nos. 3 and 4, and 9 to 14, inclusive, are fitted with superheaters.

The boilers are designed to operate the entire machinery plant at full power, with an average air pressure in the ash pits of not more than two inches of water. They are equipped primarily for burning coal, but are provided with an auxiliary installation for burning fuel oil in combination with coal.

The uptakes are of the usual design, and there are two smoke pipes, each about 92 feet high above the grates. The forward smoke pipe is 11 feet 2 inches and the after one 12 feet 9 inches inside diameter, the former serving six and the latter eight boilers. Both smoke pipes are 13 feet 5⅜ inches outside diameter.




| ITEM | DESCRIPTION | ITEM | DESCRIPTION. | ITEM | DESCRIPTION |
|---|---|---|---|---|---|
| A | 5½" BOILER STOP VALVE | I | 4½" AUX. STEAM CROSS CONNECTION | R | 9" GLOBE STOP VALVE |
| B | 5½" G.S. VALVES STEAM TO SUPERHEATER | J | 3" Do. | S | 10½" MAIN STEAM |
| C | 5½" STEAM TO SUPERHEATER | K | 5½" STEAM TO FORD. DYNAMOS | T | 10½" EXPANSION JOINT |
| D | 5½" STEAM, SUPERHEATER TO MAIN LINE | L | 4½" STEAM TO WINDLASS | U | 12" GLOBE STOP VALVE |
| E | 5½" G.S. VALVE, Do. | M | 7" MAIN STEAM CROSS CONNECTION | V | 14" MAIN STEAM |
| F | 5½" STEAM, BOILER TO MAIN LINE | N | 7" GLOBE STOP VALVE | W | 14" EXPANSION JOINT |
| G | 5½" A.S. VALVE, Do. | O | 7" EXPANSION JOINT | X | 14½" ANGLE STOP VALVE |
| H | 7" AUX. STEAM CROSS CONNECTION | P | 7" MAIN STEAM | Y | 14½" MAIN STEAM TO ENGINE ROOM |

NOTE. - STEAM PIPING ON PORT SIDE SIMILAR TO THAT SHOWN.

U.S.S. TEXAS. - ARR'G'T. OF MAIN STEAM PIPING AND MACHINERY IN BOILER ROOMS. - PLATE II.

## *Boiler Data.*

| | | |
|---|---|---|
| Number | | 14 |
| Pressure, working, pounds per square inch | | 295 |
| test, pounds per square inch | | 450 |
| Height to top, external, feet and inches | Twelve of | 12–00⅞ |
| | Two of | 11–09¾ |
| Length on floor, feet and inches | Twelve of | 10–01 |
| | Two of | 9–01½ |
| Width on floor, feet and inches | | 17–02½ |
| Drum, diameter, inside, inches | | 42 |
| length, feet and inches | | 17–09⅞ |
| thickness, inch | | 00$\frac{11}{16}$ |
| Number of furnaces, each boiler | | 1 |
| furnace doors, each boiler | | 4 |
| Grates, length, feet and inches | | 7–00 |
| width, feet and inches | | 15–10.32 |
| per cent. of air space through | | 49 |
| Total grate surface, square feet | | 1,554 |
| heating surface, generating, square feet | | 62,213 |
| superheater, square feet | | 3,267 |
| Ratio, G.S. to H.S. (generating) | | 1 to 40.03 |
| Number of tube headers, each boiler | | 27 |
| 2-inch tubes, each boiler | | 863 |
| 4-inch tubes, each boiler | | 29 |
| Distance between headers, feet and inches | Twelve of | 9–00 |
| | Two of | 8–00 |
| Area through smoke pipe, forward, square feet | | 97 |
| aft, square feet | | 127.7 |
| G.S. ÷ area through smoke pipe, forward | | 6.86 |
| aft | | 6.99 |
| Kind of forced draft | | Closed firerooms. |
| Oil burners, number each boiler | | 6 |
| type | | Peabody. |
| Diameter of, main steam stop valve, inches | | 05½ |
| main and auxiliary feed, stop and check valves (one of each), inches | | 02½ |
| surface-blow valve, inches | | 01½ |
| bottom-blow valves (two), inches | | 01½ |
| safety valve, twin, inches | | 04½ |

### FUEL—OIL SYSTEM.

In addition to the usual coal-burning equipment, a complete oil-burning system is provided, for use as auxiliary fuel to be burned as required in combination with the coal.

The plant consists of two heavy-pressure, vertical, double-acting, single, pumps,* one in each engine room on the center-line bulkhead. These pumps draw the fuel oil from the double-bottom tanks, and deliver same to the oil burners on the boilers. Each boiler is fitted with six oil burners, mechanically atomized, of the Peabody type.

A 4½-inch suction pipe leads from the bottom of each tank to the suction manifolds located near the pumps in each engine room. The pump suctions from manifolds are 5 inches in diameter each, and are each fitted with a Macomb strainer. The discharge from each pump is 3½ inches in diameter, uniting into a common 3½-inch pipe (either pump being capable of supplying the entire system) which leads to the firerooms. The 3½-inch main is fitted with a Lalor automatic stop valve at junction with discharge pipes from both pumps.

In the firerooms the main is led forward, gradually decreasing in size from 3½ to 2 inches. There is a 2-inch branch, taken off the main in each fireroom, leading to the fuel-oil heater in same fireroom. Each heater has 21.5 square feet of heating surface, the heating agent being live steam. By-passes are provided around the heaters. The heaters in firerooms Nos. 1, 3 and 4 serving four boilers each, have 1½-inch discharge connections to the burners, and that in fireroom No. 2 one of 1¼ inches, there being but two boilers in the latter compartment. Each heater discharge pipe is provided with a duplex strainer, beyond which ¾-inch branches, fitted with Lalor automatic stop valves, are led to the burners on individual boilers.

Each fuel-oil tank is provided with a 1½-inch steam connection led to the bottom of the tank, for boiling out, and a 1-inch steam fire extinguishing connection at the top of tank.

Provision is made whereby the fuel-oil pump can transfer the fuel oil, through the filling pipe, to another vessel in emergency.

For washing out the tanks, one fire and bilge pump in

*See Table I.

each engine room is arranged for pumping their contents overboard, the connection for flooding and pumping out being portable and only connected up when cleaning the tanks, so there is no danger of flooding the tanks with sea water.

## FIREROOM AUXILIARIES.

*Forced-Draft Blowers.*—Firerooms Nos. 1, 3 and 4 have four forced-draft blowers each, and fireroom No. 2 has but two. They are located in specially constructed blower rooms just below the protective deck and above the working flat in front of the boilers.

The fans are of the Sturtevant multivane type, and each is electric driven by a direct-connected Diehl motor, controlled from the fireroom working level and the blower room at will. Air is supplied from the fireroom ventilators, which are closed at the bottom when under forced draft. The blower data follows:

| | |
|---|---|
| Revolutions per minute | 825 |
| Horsepower, each | 32 |
| Fan, diameter, inches | 26½ |
| width, inches | 29⅝ |
| number of blades | 54 |

**Auxiliary Feed Pumps.*—There are four Blake, vertical, double-acting, single, auxiliary feed pumps, one in each fireroom. They are arranged so that any pump can feed any boiler.

**Fire, Bilge and Ash-Ejector Pumps.*—In each fireroom there is a Blake, vertical, double-acting, duplex pump for fire, bilge and ash-ejector service. They are arranged to draw from the drainage system and sea, and discharge to the fire main, sanitary system, ash ejectors and overboard.

*Ash Hoists.*—The port ventilator in each fireroom is fitted for hoisting ashes, the necessary bucket guides, wire ropes, sheaves, etc., being installed for the purpose. The hoists

*See Table I.

are operated from the main deck, at which deck the ash chutes are located at the ship's side.

The four ash-hoist engines, one for each hoist, are of the two-cylinder, reversible type, made by the Hyde Windlass Co. They are located in the upper fireroom hatch and are of the following principal dimensions:

| | |
|---|---|
| Number of cylinders, each | 2 |
| Diameter of cylinders, inches | 4½ |
| Stroke, inches | 4½ |

*Ash Ejectors.*—In addition to the ash hoists, there are eight 6-inch hydraulic ash expellers, two in each fireroom, port and starboard, discharging the ashes through scuppers above the protective desk.

### MAIN STEAM PIPING.

The main steam piping is arranged in two symmetrical systems, one on each side of the vessel. The two lines are cross-connected in the forward fireroom and in the engine rooms by 7- and 9-inch connections, respectively. The branches from the boilers are 5½ inches in diameter each, and the lines proper are 7 inches in the forward fireroom, increasing to 9, 10½, 12 and 14 inches at each successive boiler connection. The pipes are increased to 14½ inches through the pipe passages between the engine and firerooms, reducing again to 14 inches in the engine rooms beyond the separators, which size continues to the throttle valves.

The arrangement of valves in the main steam piping is shown in Plates I and II.

### AUXILIARY STEAM PIPING.

From the main steam pipe in the engine rooms is a 6-inch connection, with stop valve, which leads aft through the engine rooms and through the center-line bulkhead, with stop valve on either side of same, forming a connecting loop between the two sides of the ship. From this pipe steam con-

nections are taken for the various engine-room auxiliaries, steering engine and heating system aft. For use in port, when steam is shut off the main steam lines to engine rooms, there is a 4-inch auxiliary steam pipe, starboard side only, with stop valve at each end, from the auxiliary steam cross-connection in the after fireroom, which supplies steam to the auxiliary steam line in the engine rooms.

In the firerooms the auxiliary steam piping consists of a cross-connection, with stop valve at each end, between the port and starboard main steam lines, from which the branches to the auxiliaries are taken. The cross-connections are 3 inches in diameter in fireroom No. 1, 4½ inches in firerooms Nos. 2 and 3, and 7 inches in fireroom No. 4, which supplies the after dynamo room and distilling apparatus.

The forward dynamos and windlass take their steam off the main steam cross-connection in the forward fireroom.

Stop valves are fitted in all branches and sub-branches as required.

### AUXILIARY EXHAUST PIPING.

An auxiliary exhaust pipe is fitted throughout the machinery spaces and elsewhere as required for the various auxiliaries. Connections are provided to direct the exhaust steam into either main or auxiliary condensers, either feed-water heater, or into the atmosphere through the after escape pipe at will. There are also connections for admitting the exhaust steam to the L.P. receivers.

Stop valves are fitted in all branches at the main.

### MAIN AND AUXILIARY FEED SYSTEMS.

Each main feed pump has a 5½-inch independent suction from the main feed tank in same engine room. The pumps discharge *via* the feed-water heaters, or by-pass same if desired, to the boilers. The combined discharge from each pair of pumps is 6 inches, uniting in the after fireroom into an

8-inch connection leading forward and diminishing in size as it advances to 7½, 6 and 5 inches.

The auxiliary-feed suction main is taken off the 8-inch feed tanks cross-connecting pipe. This main is 9 inches in diameter, gradually reducing in size to 8, 7 and 5½ inches, as it leads forward to the auxiliary feed pumps, each pump having a 5½-inch suction connection. The auxiliary feed pumps discharge direct to the boilers in their respective compartments, or into the main feed line to any boiler. The discharge connections are 4½ inches in diameter at the pumps.

All branch main and auxiliary feed pipes to the boilers are 2½ inches in diameter.

## INTERIOR COMMUNICATION.

The customary engine and fireroom telegraphs, gongs, time-firing device, telephones, voice tubes, etc., are fitted for transmitting orders and signaling to the various machinery compartments and other parts of the vessel.

## AIR-COMPRESSOR PLANT.

Located in each engine room are four 11-inch by 11-inch by 12-inch water-cooled Westinghouse steam-driven air compressors and two air reservoir tanks of about 45,000 cubic inches capacity each, for use in running pneumatic tools in the engineering department, blowing soot off the boiler tubes and for the gas-ejecting system for the guns.

Each compressor has a capacity of about 360 cubic feet of free air per minute at 150 pounds pressure.

A pneumatic main, independent of the gun gas-ejecting system, is led throughout the machinery space, with branches to the general workshop, evaporator and dynamo rooms, from which the connections for pneumatic tools and blowing soot off boiler tubes are taken.

## EVAPORATING AND DISTILLING APPARATUS.

This plant is located on the protective deck, just forward of barbette No. 3. There are four evaporators, two distillers, two feed-water heaters, two distiller circulating pumps,* two evaporator feed pumps* and two fresh-water pumps,* together with their accessories. The plant has a combined capacity of 28,000 gallons of water per 24 hours, and is arranged to operate in double effect.

### *Evaporator Data (each).*

| | |
|---|---|
| Type | Reilly, horizontal, multi-coil. |
| Diameter, inside, feet and inches | 5–01½ |
| Length, over all, feet and inches | 5–02½ |
| Coils, number | 36 |
| diameter of pipe, inch | 01 |
| thickness of pipe, inch | 00.065 |
| Heating surface, square feet | 221.4 |
| Diameter of steam connection, inches | 03 |
| vapor nozzle, inches | 04½ |
| feed valve, inches | 01¼ |
| blow valve, inches | 01½ |

### *Distiller Data (each).*

| | |
|---|---|
| Type | Reilly, vertical, multi-coil. |
| Diameter, inside, inches | 31 |
| Length, over all, feet and inches | 4–03⅜ |
| Coils, number | 11 |
| diameter, inch | 01 |
| thickness, inch | 00.065 |
| Cooling surface, square feet | 92.25 |
| Diameter of circulating water inlet and outlet, inches | 04½ |
| vapor inlet, inches | 04½ |
| drain, inches | 02 |

### *Evaporator Feed-Water Heater Data (each).*

| | |
|---|---|
| Type | Reilly, vertical, multi-coil. |
| Diameter, inside, inches | 12 |
| Length, over all, inches | 48¾ |

*See Table I.

| | |
|---|---|
| Coils, number | 2 |
| diameter, inch | 01 |
| thickness, inch | 00.065 |
| Heating surface, square feet | 12.3 |
| Diameter of feed inlet and outlet, inches | 02 |
| vapor inlet and outlet, inches | 04½ |

GENERAL WORKSHOP.

A well-equipped machine shop is located amidship, on the protective deck, between the engine-room hatches.

The following machine tools, each driven by its own electric motor and all up to date and complete with the most modern attachments, are installed as listed below:

| No. | Description. | Make and H.P. of Motor. |
|---|---|---|
| 1 | 28-inch by 48-inch swing, extension-gap lathe, 10 feet between centers; The Rahn Larmon Co., Cincinnati, Ohio.. | Diehl, 5 H.P. |
| 1 | 14-inch swing lathe, 40 inches between centers; The American Tool Works Co., Cincinnati, Ohio..... | Diehl, 2 H.P. |
| 1 | 14-inch swing lathe, 5 feet 4 inches between centers; The American Tool Works Co., Cincinnati, Ohio..... | Diehl, 2 H.P. |
| 1 | Column shaper, 15-inch stroke by 15-inch traverse; The American Tool Works Co., Cincinnati, Ohio..... | Diehl, 2 H.P. |
| 1 | 30-inch radial drill, capacity two-inch holes; The American Tool Works Co., Cincinnati, Ohio..... | Diehl, 3 H.P. |
| 1 | 12-inch sensitive drill, capacity ⅜-inch holes; Jas. Clark, Jr., Electric Co., Louisville, Ky. | |
| 1 | Universal milling machine, 24½-inch longitudinal feed, 18-inch vertical movement and 7½-inch traverse; The Kempsmith Mfg. Co., Milwaukee, Wis. | Diehl, 2 H.P. |

1 Double emery grinder on column; Ransom Mfg. Co., Oshkosh, Wis..... G. E., 3 H.P.
1 Portable cylinder-boring machine, capacity 6 inches to 24 inches with 30-inch travel; H. B. Underwood and Co., Philadelphia, Pa..... Diehl, 2 H.P.

BLACKSMITH SHOP.

A blacksmith shop is provided on the main deck, in the deck house, abreast of the after smoke pipe, port side. It is equipped with one portable and one permanent forge, together with anvil and all necessary tools and fittings. The permanent forge is fitted with an electrically-driven blast fan.

FOUNDRY.

A small foundry is installed in the deck house on the main deck, frames 64–66, amidship. The foundry outfit consists of a small oil-burning crucible furnace of the Bureau's standard type, together with adequate allowance of crucibles and the necessary apparatus for handling and pouring the metal.

ELECTRIC PLANT.

There are two dynamo rooms, one just forward of the forward boiler room and the other just aft of the after boiler room. They are on the lower platform level, with their condensing apparatus below in a separate room in the hold, there being an access hatch and ladder between the two levels.

The distributions rooms, two in number, are located on the upper platform, one over each dynamo room. They contain the lighting, power and searchlight distribution boards only. The generator boards are in the dynamo rooms.

The generator installation consists of four 6-pole, compound-wound, 300-kilowatt, General Electric generators, two in each dynamo room, each driven by a two-stage horizontal

Curtis turbine. Each generator will deliver at normal load 2,400 ampères of current at 125 volts, when running at 1,500 revolutions per minute. The generators are capable of delivering one-third overload for two hours without injury.

There is one condenser for each pair of generators, of the same general design as the main and auxiliary condensers. Each condenser has its independent air pump,* centrifugal circulating pump,* hotwell-tank pump,* and a hotwell tank.

*Dynamo Condenser Data.*

| | |
|---|---|
| Number | 2 |
| Inside diameter, feet and inches | 4–05½ |
| Thickness of shell (steel), inch | 00⅜ |
| Length between tube sheets, feet and inches | 8–05 |
| Thickness of tube sheets, inches | 01⅛ and 01⅜ |
| Tubes, number | 1,745 |
| diameter, outside, inch | 00⅝ |
| thickness, inch | 00.065 |
| Cooling surface, square feet | 2,402 |
| Diameter of exhaust nozzles (2), inches | 20 |
| air-pump suctions, inches | 07 |
| circulating-water inlet and outlet, inches | 08 |

*Dynamo-Condenser Air Pumps Data.*

| | |
|---|---|
| Number | 2 |
| Type | Blake, vertical, twin, beam, single steam cylinder. |
| Diameter of suction nozzle, inches | 07 |
| discharge nozzle, inches | 06 |
| steam cylinder (1), inches | 09 |
| water cylinders (2), inches | 18 |
| Stroke, inches | 12 |

*Dynamo-Condenser Circulating Pumps and Engines Data.*

| | |
|---|---|
| Number | 2 |
| Type | Centrifugal, driven by single engine. |
| Diameter of impeller, inches | 26 |
| suction (2), inches | 05¾ |
| discharge, inches | 08 |
| steam cylinder, inches | 06 |
| Stroke, inches | 05 |

*See Table I.

## TRIALS.

The contract required four trials as follows:

(*a*) A progressive trial over a measured-mile course in water not less than 40 fathoms deep for standardizing the screws, extending from maximum speed down to a speed of ten knots; about twenty-three runs to be made over the course in order to adequately cover the range of speed desired.

(*b*) A full-speed trial of four hours' duration in the open sea in deep water, at the highest speed attainable, the speed developed not to be less than an average of 21 knots. During this trial the air pressure in the firerooms shall not exceed an average of 2 inches of water, and the steam pressure in the H.P. steam chest shall not exceed 265 pounds above the atmosphere.

(*c*) An endurance and coal-and-water-consumption trial of twenty-four hours in the open sea in deep water at a speed as nearly uniform as possible, and not less than an average of 19 knots. All necessary auxiliaries shall be in operation, including all those usually required under service cruising conditions, the distilling plant excluded.

(*d*) An endurance and coal-and-water-consumption trial of twenty-four hours in the open sea at an average uniform speed of 12 knots as nearly as possible, under conditions similar to Trial (*c*).

(*e*) A trial of two hours' duration at the highest speed attainable, burning coal and oil fuel in combination.

The fuel consumption was carefully measured on Trials (*b*), (*c*), (*d*) and (*e*), and the water consumption on trials (*b*), (*c*) and (*d*); the contractors guaranteeing that the water consumption, including water used in all auxiliaries in use on the trial, except water used and made in the distilling plant, would not exceed 20,000 pounds, 14,210 pounds and 7,420 pounds per knot run, for trials (*b*), (*c*) and (*d*), respectively.

*Standardization Trial (a).*—This trial was run on the measured-mile course off Rockland, Maine. The trial was com-

menced on October 23, 1913, but owing to an accident to the forward L.P. valve gear of the port engine, which occurred on the twentieth run while making approximately 21 knots, the trial was brought to an abrupt termination. The vessel steamed slowly back to anchor under the starboard engine alone, and repairs were immediately begun, which required three days to complete. A description of the accident was published on pages 767 and 768, of Vol. XXV, of the JOURNAL, to which the reader is referred.

Upon completion of repairs the trial was resumed October 27, at which time all the high-speed runs were made. The contractors not being satisfied with the 19-knot point obtained on the first day, October 23, believing the vessel had not acquired full acceleration, repeated these runs on the morning of October 28, allowing the vessel a better start in which to gather momentum, and obtained a slightly improved point for the speed curve.

In all thirty-four runs were made over the measured-mile at various speeds, but the data from only twenty-three was used in plotting the speed and power curves, runs Nos. 13 to 21, inclusive, and No. 30, not being up to the speed desired, were thrown out.

From the data obtained it was found to require 124 r.p.m. of the main engines to attain the designed speed of 21 knots, 109.85 r.p.m. for 19 knots and 68.45 r.p.m. for 12 knots.

Table II contains the data obtained on the various runs, from which the curves, Plate III, were plotted.

*Four-Hour Full-Speed Trial (b).*—At 10:40 A. M., October 28, 1913, the four-hour full-speed trial was begun. It took place at sea off Rockland, Maine, and was successfully concluded at 2:40 P. M. The weather was overcast and hazy, with a light breeze and smooth sea. The designed speed was readily attained without effort on the part of the engine-room and fireroom forces. The average speed for the four hours was 21.05 knots, and the guaranteed water rate was just met.

TABLE II.

STANDARDIZATION TRIAL DATA.—U.S.S. TEXAS.—COURSE OFF ROCKLAND, MAINE, OCTOBER 23, 27 & 28, 1913.

| No. of Run | Time on Course | | Speed in Knots | Revs. to go One Knot | | | R.P.M. | | | I.H.P. | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Mins. | Secs. | | Starbd. Engine | Port Engine | Mean | Starbd. Engine | Port Engine | Mean | Starbd. Engine | Port Engine | Total |
| 1 | 5 | 39.5 | 10.604 | 323.1 | 322.7 | 322.9 | 57.13 | 57.07 | 57.10 | 1,267 | 1,471 | 2,738 |
| 2 | 6 | 21.8 | 9.429 | 367.3 | 359.2 | 363.1 | 57.67 | 56.38 | 57.02 | 1,270 | 1,403 | 2,673 |
| 3 | 5 | 41.9 | 10.529 | 326.5 | 324.5 | 325.5 | 57.30 | 56.96 | 57.13 | 1,289 | 1,382 | 2,671 |
| Mean of Group | | | 9.998 | | | | | | 57.07 | | | 2,689 |
| 4 | 5 | 7.2 | 11.719 | 353.0 | 354.6 | 354.8 | 68.97 | 69.67 | 69.32 | 2,152 | 3,017 | 5,169 |
| 5 | 4 | 45.5 | 12.609 | 328.9 | 328.5 | 328.7 | 69.14 | 69.05 | 69.10 | 2,309 | 2,230 | 4,539 |
| 6 | 5 | 10.6 | 11.590 | 357.7 | 359.5 | 358.6 | 69.10 | 69.46 | 69.28 | 2,305 | 2,154 | 4,459 |
| Mean of Group | | | 12.132 | | | | | | 69.20 | | | 4,677 |
| 7 | 3 | 53.4 | 15.424 | 333.4 | 329.8 | 331.6 | 85.75 | 84.80 | 85.28 | 4,535 | 4,065 | 8,600 |
| 8 | 4 | 8.6 | 14.481 | 353.7 | 350.9 | 352.3 | 85.96 | 84.69 | 85.03 | 4,457 | 3,908 | 8,365 |
| 9 | 3 | 57.6 | 15.151 | 3350 | 332.7 | 333.9 | 84.54 | 84.01 | 84.28 | 4,402 | 3,899 | 8,301 |
| Mean of Group | | | 14.884 | | | | | | 84.91 | | | 8,408 |
| 10 | 3 | 43.1 | 16.196 | 355.4 | 353.4 | 354.4 | 95.59 | 95.06 | 95.33 | 6,083 | 5,650 | 11,733 |
| 11 | 3 | 32.3 | 16.957 | 341.8 | 338.8 | 340.3 | 96.52 | 95.74 | 96.13 | 6,407 | 5,845 | 12,252 |
| 12 | 3 | 40.2 | 16.349 | 353.1 | 351.4 | 352.3 | 96.18 | 95.73 | 95.96 | 6,340 | 5,826 | 12,166 |
| Mean of Group | | | 16.600 | | | | | | 95.89 | | | 12,101 |
| 22 | 2 | 47.0 | 21.557 | 337.9 | 338.5 | 338.2 | 121.44 | 121.66 | 121.55 | 12,964 | 12,747 | 25,711 |
| 23 | 3 | 0.9 | 19.901 | 366.7 | 369.2 | 368.0 | 121.60 | 122.42 | 122.01 | 13,459 | 13,299 | 26,758 |
| 24 | 2 | 45.2 | 21.792 | 337.0 | 336.6 | 336.8 | 122.44 | 122.31 | 122.38 | 13,227 | 13,668 | 26,895 |
| Mean of Group | | | 20.788 | | | | | | 121.99 | | | 26,631 |
| 25 | 2 | 59.4 | 20.067 | 372.6 | 373.8 | 373.2 | 124.60 | 125.02 | 124.81 | 14,436 | 14,510 | 28,946 |
| 26 | 2 | 42.7 | 22.127 | 340.8 | 340.8 | 340.8 | 125.55 | 125.55 | 125.55 | 14,156 | 14,426 | 28,582 |
| 27 | 2 | 59.7 | 20.033 | 373.9 | 375.0 | 374.5 | 124.84 | 125.21 | 125.03 | 14,229 | 14,273 | 28,502 |
| 28 | 2 | 41.6 | 22.277 | 339.7 | 341.2 | 340.4 | 126.10 | 126.68 | 126.39 | 14,572 | 14,936 | 29,508 |
| 29 | 2 | 59.3 | 20.078 | 371.0 | 375.7 | 373.4 | 124.21 | 125.80 | 125.01 | 13,970 | 14,702 | 28,672 |
| Mean of Group | | | 21.127 | | | | | | 125.47 | | | 28,850 |
| 31 | 3 | 12.1 | 18.740 | 348.3 | 351.9 | 350.0 | 108.76 | 109.87 | 109.32 | 8,672 | 9,053 | 17,725 |
| 32 | 3 | 11.8 | 18.770 | 344.2 | 349.7 | 347.0 | 107.65 | 109.37 | 108.51 | 8,812 | 8,873 | 17,685 |
| 33 | 3 | 9.8 | 18.967 | 342.7 | 347.8 | 345.2 | 108.37 | 109.98 | 109.18 | 8,767 | 8,945 | 17,712 |
| Mean of Group | | | 18.812 | | | | | | 108.88 | | | 17,702 |

Table III gives a comparative list of the data obtained on this and the following trials.

Plate IV shows indicator cards from set No. 3, taken on this trial, and Plate V the combined card of same set.

*Twenty-Four-Hour 12-Knot Endurance and Coal-and-Water-Consumption Trial (d).*—This trial followed the full-speed trial on October 28 and 29, 1913. It was run during the twenty-four hours commencing 9:30 P. M. The weather was overcast and foggy throughout most of the trial, but otherwise very good. The average speed was 12.137 knots, just in excess of that desired, and the water consumption guarantee was appreciably bettered.

For data see Table III.

*Twenty-Four-Hour 19-Knot Endurance and Coal-and-Water-Consumption Trial (c).*—The trial began at 1:05 A. M.,

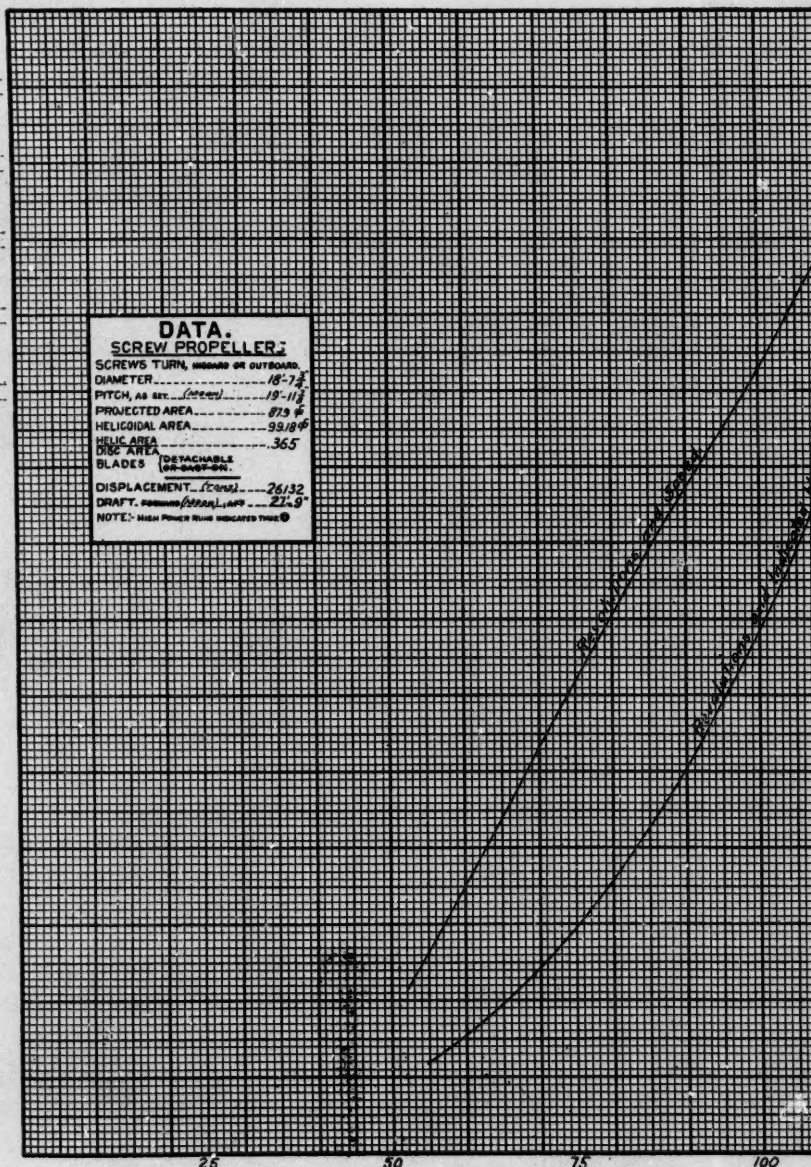
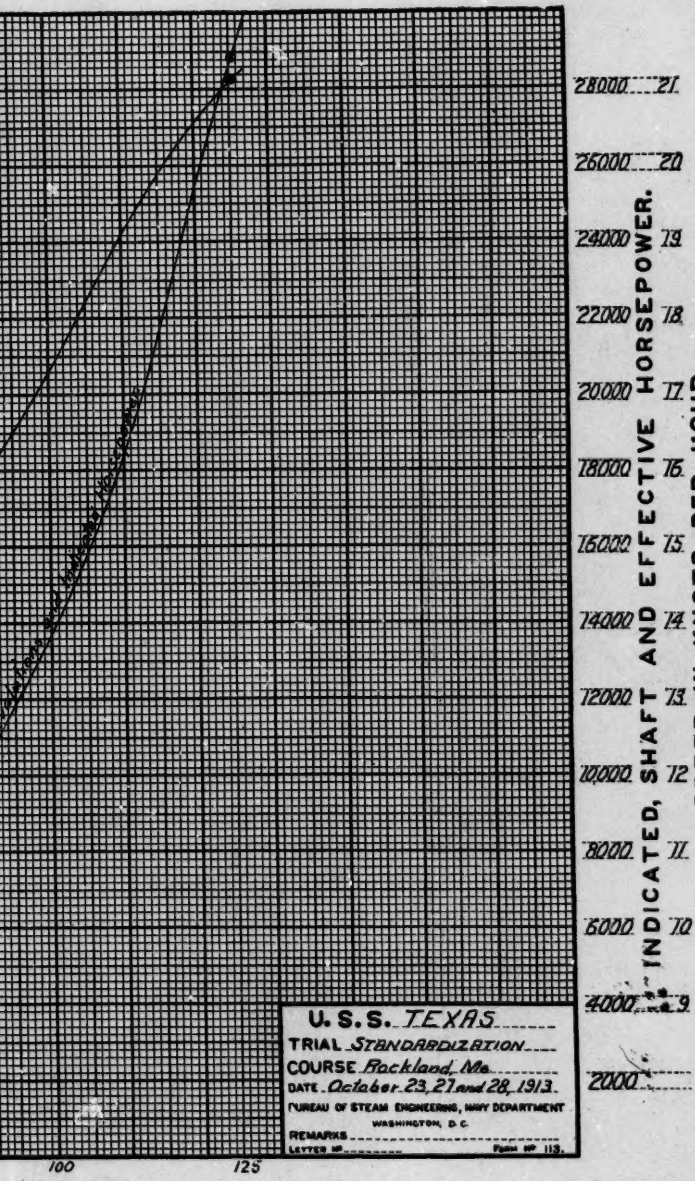

PROPULSIVE COEFFICIENT
DATA.
SCREW PROPELLERS
SCREWS TURN, INBOARD OR OUTBOARD.
DIAMETER 18'-7 3/4"
PITCH, AS SET. (mean) 19'-11 1/2"
PROJECTED AREA 879 sq ft
HELICOIDAL AREA 991.8 sq ft
HELIC AREA .365
DISC AREA
BLADES (DETACHABLE OR CAST-ON.
DISPLACEMENT (tons) 26132
DRAFT. FORWARD (mean) AFT 27'-9"
NOTE:- HIGH POWER RUNS INDICATED THUS ⊕
25
50
75
100
REVOLUTIONS PER MINUT

PLATE III.


28000 21
26000 20
24000 19
22000 18
20000 17
18000 16
16000 15
14000 14
12000 13
10000 12
8000 11
6000 10
4000 9
2000
INDICATED, SHAFT AND EFFECTIVE HORSEPOWER.
SPEED IN KNOTS PER HOUR.
U.S.S. TEXAS
TRIAL STANDARDIZATION
COURSE Rockland, Me.
DATE October 23, 27 and 28, 1913.
BUREAU OF STEAM ENGINEERING, NAVY DEPARTMENT
WASHINGTON, D.C.
REMARKS
LETTER NO.
FORM NO. 113.
100
125

MINUTE.

# TABLE III
## U.S.S. TEXAS - TRIAL DATA.

| | 4 HOURS' TRIAL (b) | 24 HRS.' 19 KTS. TRIAL (c) | 24 HRS.' 12 KTS. TRIAL (d) | 2 HOURS' COAL & OIL BURNING TRIAL (e) |
|---|---|---|---|---|
| DATE OF TRIAL | OCT. 28, 1913 | OCT. 30 & 31, 1913 | OCT. 28 & 29, 1913 | OCT. 31, 1913 |
| SPEED IN KNOTS | 21.05 | 19.078 | 12.137 | 20.988 |
| DRAUGHT, MEAN ON TRIAL, FEET AND INCHES | 27 - 9¼ | 27 - 9 | 27 - 8 | |
| DISPLACEMENT, CORRESPONDING, TONS | 26,250 | 26,230 | 26,130 | |
| PRESSURES (AVERAGE): | STARBD. \| PORT | STARBD. \| PORT | STARBD. \| PORT | STARB'D. \| PORT |
| MAIN STEAM, AT BOILERS, POUNDS GAUGE | 295.6 | 284.9 | 285.5 | 287.4 |
| ENGINES, DO. | 274.1 \| 274.1 | 276.5 \| 276.5 | 274.8 \| 279.5 | 265.2 \| 262.2 |
| 1ST REC'S., POUNDS ABSOLUTE | 94.1 \| 88.3 | 75.6 \| 71.1 | 32.9 \| 31.6 | |
| 2ND REC'S., DO. | 27.9 \| 25.06 | 19.2 \| 18.3 | 11.6 \| 8.09 | |
| VACUUM, MAIN CONDENSERS, INCHES OF MERCURY | 26.72 \| 26.9 | 26.71 \| 26.92 | 26.34 \| 27.07 | |
| BAROMETER, INCHES OF MERCURY | 29.98 | 30.06 | 29.94 | |
| MAIN FEED, AT PUMPS, POUNDS GAUGE | 391.3 | 352.3 | 345.8 | |
| AUXILIARY EXHAUST, DO. | 12.5 | 11.3 | 0.5" VACUUM | |
| FORCED LUBRICATION, DO. | 51.5 | 52.6 | 46.5 | |
| FUEL OIL, DO. | NOT USED | NOT USED | NOT USED | 150 |
| STEAM TO FUEL OIL HEATERS, POUND GAUGE | DO. | DO. | DO. | 22.5 |
| AIR PRESSURE IN FIRE ROOMS, INCHES OF WATER | 1.48 | 0.275 | 0 | 1.91 |
| MEAN EFFECTIVE PRESSURES, POUNDS (AVERAGE): | | | | |
| H.P. CYLINDER | 120.2 \| 132.3 | 90.8 \| 96 | 36.10 \| 41.35 | |
| I.P. DO. | 45.9 \| 47.8 | 38.1 \| 36 | 15.10 \| 13.77 | |
| F.L.P. DO. | 17.1 \| 15.2 | 12.25 \| 11.43 | 4.62 \| 4.35 | |
| A.L.P. DO. | 18.04 \| 15.5 | 12.04 \| 11.45 | 4.63 \| 4.15 | |
| REFERRED PRESSURE | 43.79 \| 43.39 | 32.96 \| 32.32 | 12.88 \| 12.71 | |
| REVOLUTIONS OR DOUBLE STROKES PER MINUTE (AVERAGE): | | | | |
| STARBOARD ENGINE | 124.35 | 110.27 | 69.16 | 123.67 |
| PORT ENGINE | 124.76 | 110.32 | 69.36 | 124.17 |
| MEAN | 124.56 | 110.30 | 69.26 | 123.92 |
| MAIN AIR PUMPS | 28.9 \| 24.3 | 21.1 \| 26.9 | NOT USED \| 30.7 | |
| CIRCULATING PUMPS | 237 \| 219.8 | 193.3 \| 191.4 | 152.6 \| 154.1 | |
| FEED PUMPS | (4) 22.4 | (4) 12.8 | (1) 17.7 | |
| FORCED DRAFT BLOWERS | 734.3 | 657.5 | NOT USED | |
| FUEL OIL PUMPS | NOT USED | NOT USED | NOT USED | 8 |
| INDICATED HORSEPOWER (AVERAGE): | | | | |
| H.P. CYLINDER | 4,236 \| 4,686 | 2,840 \| 3,014 | 711 \| 815 | |
| I.P. DO. | 4,275 \| 4,468 | 3,150 \| 2,971 | 788 \| 716 | |
| F.L.P. DO. | 2,775 \| 2,479 | 1,764 \| 1,649 | 420 \| 394 | |
| A.L.P. DO. | 2,927 \| 2,527 | 1,735 \| 1,656 | 420 \| 376 | |
| TOTAL | 14,213 \| 14,160 | 9,489 \| 9,290 | 2,339 \| 2,301 | |
| TOTAL BOTH ENGINES | 28,373 | 18,779 | 4,640 | |
| AUXILIARIES (ESTIMATED FROM CURVES) | 1,025 | 805 | 430 | |
| TOTAL ALL MACHINERY | 29,398 | 19,584 | 5,070 | |
| TEMPERATURES, DEGREES F (AVERAGE): | | | | |
| MAIN STEAM, AT ENGINES | 451.7 \| 457 | 452.9 \| 447.8 | 450.5 \| 449.6 | |
| STEAM, 1ST RECEIVERS | 325 \| 319 | 310.9 \| 303.5 | 252.2 \| 246.2 | |
| 2ND DO. | 245 \| 239.6 | 225.5 \| 218.3 | 187.4 \| 183.7 | |

DATA TAKEN.

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| AUXILIARIES (ESTIMATED FROM CURVES) | 1,025 | | 805 | | 430 | | | |
| TOTAL ALL MACHINERY | 29,398 | | 19,584 | | 5,070 | | | |
| TEMPERATURES, DEGREES F (AVERAGE): | | | | | | | | |
| MAIN STEAM, AT ENGINES | 451.7 | 457 | 452.9 | 447.8 | 450.5 | 449.6 | | |
| STEAM, 1ST RECEIVERS | 325 | 319 | 310.9 | 303.5 | 252.2 | 246.2 | | |
| 2ND DO. | 245 | 239.6 | 225.5 | 218.3 | 187.4 | 183.7 | | |
| SUPERHEAT, MAIN STEAM, AT ENGINES | 37.7 | 43 | 38.1 | 33 | 36.3 | 33.9 | | |
| STEAM, 1ST RECEIVERS | 1.5 | 0 | 1.8 | 0 | 0 | 0 | | |
| 2ND DO. | 0 | 0 | 0 | 0 | 0 | 0 | | |
| MAIN INJECTION | 53.5 | | 60.85 | | 55.8 | | | |
| OVERBOARD DISCHARGE | 96.5 | | 95 | | 72.1 | | | |
| AIR PUMPS DISCHARGE | 90.4 | | 91.5 | | 72.2 | | | |
| FEED WATER | 170.6 | | 92.9 | | 91.7 | | | |
| ENGINE ROOMS, WORKING PLATFORM | 92.1 | | 89.3 | | 82 | | | |
| FIRE ROOMS, DO. | 105.6 | | 116.2 | | 113.2 | | | |
| OUTSIDE AIR | 57.5 | | 55.6 | | 59 | | | |
| SMOKE PIPES | №1,594; №2,616 | | №1,400; №2,610 | | №2,352.3 | | | |
| FUEL OIL, TO HEATERS | NOT USED | | NOT USED | | NOT USED | | | 66 |
| BURNERS | DO. | | DO. | | DO. | | | 146 |
| WATER CONSUMPTION DATA: | | | | | | | | |
| POUNDS PER HOUR | 427,182 | | 271,727 | | 80,579 | | | |
| PER I.H.P. MAIN ENGINES | 15.06 | | 14.47 | | 17.37 | | | |
| ALL MACHINERY | 14.53 | | 13.87 | | 15.89 | | | |
| KNOT RUN (GUARANTEE) | AT 20,000 | | AT 14,210 | | AT 7,420 | | | |
| (TRIAL PERFORMANCE) | 21-KTS. 20,000 | | 19-KTS. 14,095 | | 12-KTS. 6,482 | | | |
| (BELOW GUARANTEE) | 0 | | 115 | | 938 | | | |
| FUEL CONSUMPTION DATA: | | | | | | | | |
| COAL, KIND | NEW RIVER HAND PICKED | | NEW RIVER HAND PICKED | | NEW RIVER HAND PICKED | | | NEW RIVER HAND PICKED |
| QUALITY | | | | | | | | |
| B.T.U. PER POUND | 14,970 | | 14,970 | | 14,970 | | | 14,970 |
| POUNDS PER KNOT RUN | 2,332 | | 1,508 | | 768 | | | 1,562 |
| HOUR | 49,084 | | 28,773 | | 9,324 | | | 32,785 |
| PER I.H.P., MAIN ENGINES | 1.73 | | 1.53 | | 2.01 | | | |
| ALL MACHINERY | 1.67 | | 1.47 | | 1.84 | | | |
| SQ. FT. OF GRATE | 36.85 | | 21.60 | | 14 | | | 21.1 |
| H.S. | 0.789 | | 0.505 | | 0.344 | | | 0.527 |
| FUEL OIL, B.T.U. PER POUND | NOT USED | | NOT USED | | NOT USED | | | 19,330 |
| POUNDS PER HOUR | DO. | | DO. | | DO. | | | 15,476 |
| KNOT RUN | DO. | | DO. | | DO. | | | 737 |
| POUND OF COAL | DO. | | DO. | | DO. | | | 0.472 |
| MISCELLANEOUS DATA: | | | | | | | | |
| SLIP OF PROPELLERS, PER CENT. OWN SPEED (MEAN) | 14.19 | 14.47 | 12.30 | 12.34 | 11.05 | 11.90 | | |
| NUMBER OF BOILERS IN USE | .14 | | 10-HRS., 14; 14-HRS., 12 | | 6 | | | 14 |
| GRATE SURFACE USED, SQ. FT. | 1,554 | | 1,424.5 AVERAGE | | 666 | | | 1,554 |
| HEATING SURFACE USED, SQ. FT. (GENERATING) | 62,213 | | 56,946 | | 27,087 | | | 62,213 |
| I.H.P. PER SQ. FT. OF GRATE, MAIN ENGINES | 18.26 | | 13.18 | | 6.97 | | | |
| ALL MACHINERY | 18.92 | | 13.75 | | 7.61 | | | |
| SQ. FT. OF H.S. PER I.H.P. MAIN ENGINES | 2.192 | | 3.052 | | 5.838 | | | |
| ALL MACHINERY | 2.116 | | 2.907 | | 5.343 | | | |
| C.S. (MAIN COND.) PER I.H.P. MAIN ENGINES | 0.924 | | 1.396 | | 5.648 | | | |
| ALL MACH'Y. | 0.891 | | 1.338 | | 5.169 | | | |
| KNOTS PER TON OF COAL | 0.961 | | 1.486 | | 2.911 | | | |
| CUTOFF: | | | | | | | | |
| H.P. CYLINDER | .771 | .80 | .611 | .601 | .54 | .54 | | |
| I.P. DO. | .78 | .777 | .665 | .665 | .633 | .639 | | |
| F.L.P. DO. | .62 | .62 | .62 | .62 | .536 | .54 | | |
| A.L.P. DO. | .62 | .62 | .62 | .62 | .538 | .54 | | |

NO OTHER DATA TAKEN.

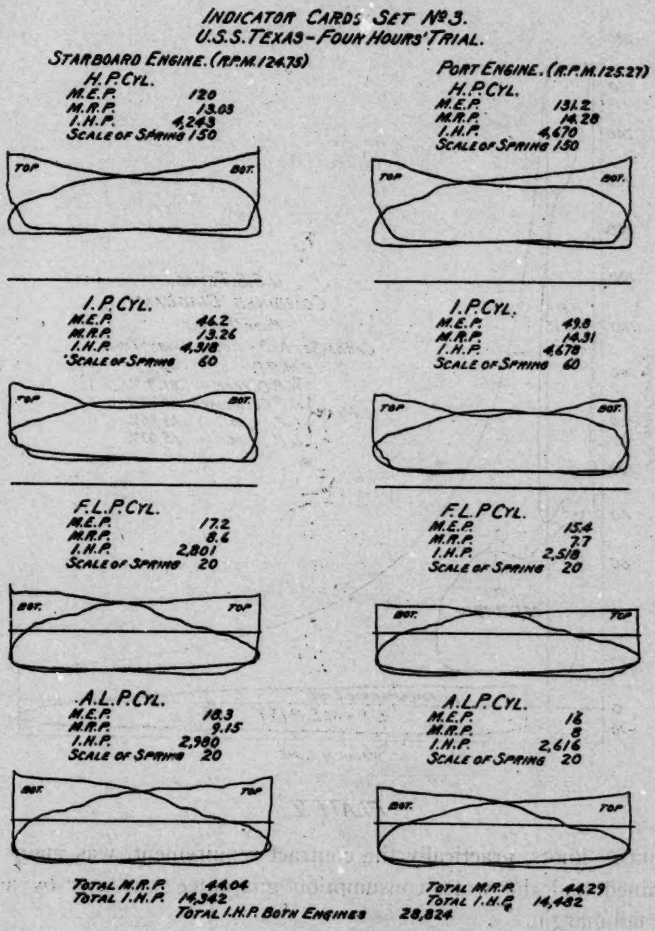



PLATE IV.

October 30, 1913, and terminated at the same hour on the following day. The weather conditions were very good, a fresh breeze and moderate sea, which died down in the afternoon of October 30, and during the remainder of the trial the breezes were light and variable. An average speed of

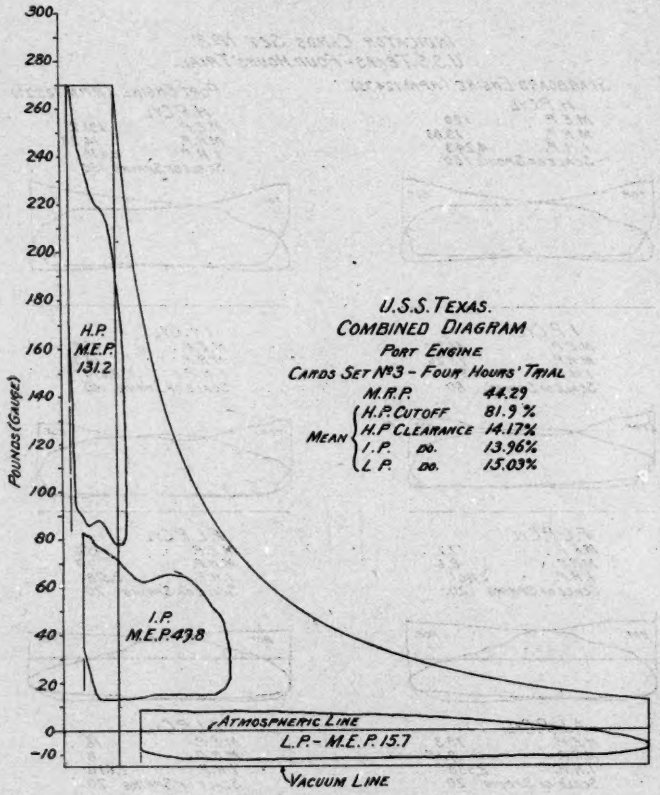



PLATE V

19.078 knots, practically the contract requirement, was maintained and the water-consumption guarantee was met by a small margin.

For data see Table III.

*Two-Hour Coal and Oil-Burning Trial (e).*—Following the 19-knot trial, this trial was started at 9:20 A. M., October 31, 1913, and successfully concluded two hours later off the Virginia Capes. Fresh breezes were encountered, but the sea was smooth.

For data see Table III.

The operation and handling of the machinery throughout the trials, and the expeditious and skillful manner in which the wrecked valve gear was repaired in incredible time and in the face of many obstacles, reflect much credit upon the builders; particularly the engineer's organization, which under the leadership of the Superintendent of Machinery, Mr. N. Christiansen, and his able assistant, Mr. Jas. Williamson, showed a unity of action and attention to duty that could not easily be excelled.

# CHINESE CRUISER *FEI HUNG*, BUILT BY NEW YORK SHIPBUILDING COMPANY, CAMDEN, NEW JERSEY.

## DESCRIPTION AND TRIALS, JANUARY, 1914.

BY ERNEST H. RIGG, ASSOCIATE.

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### GENERAL DESCRIPTION OF THE VESSEL.

The *Fei Hung* is a light cruiser, built for the Chinese Navy for use primarily as a training ship; she is also available for all the war functions that can be expected from a vessel of this size and type. She is a deck-protected cruiser, driven by turbine machinery and carrying an armament fully up to the standard of her class.

The *Fei Hung* was built by the New York Shipbuilding Company of Camden, New Jersey, from outline requirements supplied by the Chinese Government; she is of the same general type as the *Ying Swei* and *Chao Ho*, recently completed in England. Some latitude was allowed the builders, so that all these ships have minor differences in arrangement and motive power, notably in the matter of boiler power. What follows relates only to the *Fei Hung* unless specially noted.

The outline data of the hull is as follows:

| | |
|---|---|
| Length overall, feet and inches | 322–00 |
| Length between perpendiculars, feet and inches | 320–00 |
| Beam, molded, feet and inches | 39–00 |
| Depth, feet and inches | 22–06 |
| Normal displacement, tons | 2,600 |
| Normal draught, feet and inches | 14–00 |
| Freeboard forward, feet and inches | 18–03 |
| Normal complement, total | 230 |
| Bunker capacity, coal, tons | 600 |
| oil fuel, tons | 100 |
| Full forecastle, length, feet and inches | 87–00 |

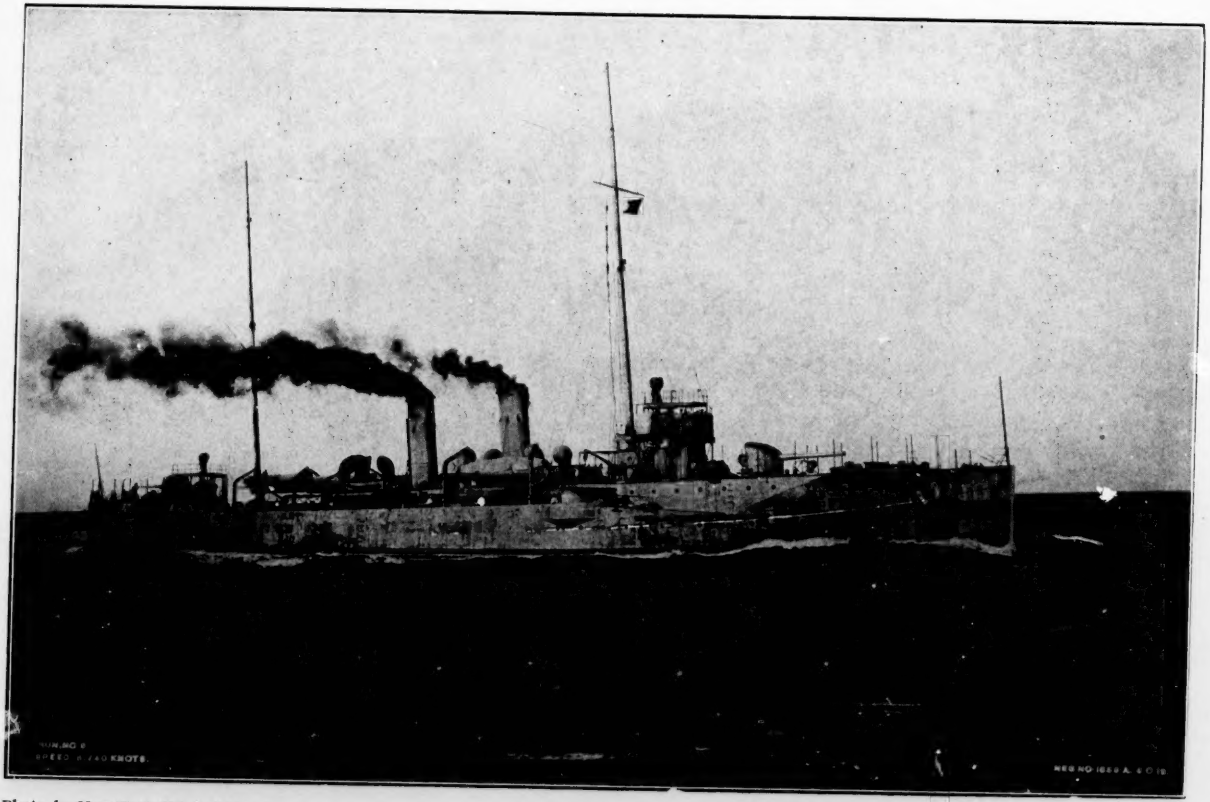


Photo. by New York Shipbuilding Co.

CHINESE CRUISER "FEI HUNG" ON TRIAL TRIP.

| | |
|---|---|
| Full poop, length, feet and inches | 58–00 |
| Double bottom, under machinery, feet and inches | 117–00 |
| Deck protection, on flat, inches | 01 |
| slopes, inches | 02 |

Bunkers (coal) above and below the protective deck, amidships.

The accompanying photo shows the general appearance of the vessel; the officers are berthed in fifteen staterooms forward in the forecastle and on deck; the crew are in large compartments on the berth deck and in the poop. A large sick bay is fitted in the poop. Galleys are on deck amidships. Magazines are located in a platform deck, both forward and aft, with storerooms, etc., below in the holds, thus ensuring a maximum of protection. The oil fuel is carried in suitable compartments at each end of the machinery spaces.

Trimming tanks, potable water, reserve feed tanks and water-ballast tanks are provided.

Cold-storage compartments are fitted on the protective deck abreast the engine hatch. Stowage for torpedoes is provided below deck.

The weather decks are wood sheathed. Under the bridge an armored conning tower and communication tube are fitted.

A very complete outfit of boats is provided, including one steam launch and one motor whaleboat; there are eight boats and rafts in all. Anchors, chain and lines are provided as customary for vessels of this class.

Dismounting bogies, auxiliary davits, etc., are provided for handling guns, ammunition and stores.

The ship is fitted with a wireless-telegraphic outfit, the masts being of extra height to carry the aerials.

## ARMAMENT.

The vessel is armed as follows:

Two 6-inch guns, 50 calibers, with shields.
Four 4-inch guns, 50 calibers, with shields.
Two 3-inch guns.
Six 3-pounders.
Two 37-millimeter automatics.
Two 18-inch deck torpedo tubes.
Small arms for crew.

The 6-inch guns are located one on the forecastle and one on the poop; the 4-inch are on the main deck at the breaks of poop and forecastle; the 3-inch, 3-pounders and torpedo tubes are arranged along the waist, and the automatics on the forecastle deck under the bridge. All this armament, including the first supply of ammunition, was supplied by Sir W. G. Armstrong, Whitworth & Company, of England, in order that the armament of all the ships should be of similar make, using the same ammunition. The automatics are by Vickers, Limited. The gun trials were successful in every respect. Several rounds per gun were fired with full charges at varying degrees of train, followed by a broadside; this trial was carried out in connection with the speed trials. Armored hoists, electrically operated by 3-horsepower motors, are fitted to each 6-inch gun; the other guns being served by hand whips. Torpedoes are stowed in boxes on deck and below in a special space. A dummy torpedo was fired from each tube both by compressed air and by a powder charge whilst at the builder's works. On the trials, an actual torpedo, with practice head, was fired by air.

## PROPELLING MACHINERY.

This vessel is fitted with an installation of Parsons turbines of the regular standard all-reaction type and design, which has proven so successful in vessels of this type. The machinery is arranged in one engine room, and there are three lines of shafting, with one propeller on each shaft.

The high-pressure ahead turbine drives the center shaft and has fifty-six rows of blades in the casing, and a similar number on the rotor, whilst the diameter of the rotor is 39 inches. At the forward end of this turbine a cruising element is arranged, which has an additional sixteen rows of blades in casing and rotor respectively. This stage helps to improve the economy of the turbines at cruising speeds.

A combined L.P. ahead and astern turbine drives each wing shaft, the L.P. ahead turbines having sixty-four rows of blades in casing and rotor respectively, with a drum diameter of 55

inches. Each astern turbine has forty rows of blades in casing and rotor respectively, drum diameter being 44 inches.

The turbine bearings are supplied with oil under pressure, whilst the line-shaft bearings are lubricated on a ring system.

The turbine machinery worked most satisfactorily throughout all the trials, and the turbine-bearing journals retained a constant temperature at all times. Heavy weather was experienced between the runs, with excessive rolling of the ship. but this did not appear to affect the steady running of the turbines either at high or low speeds.

Dummy readings were taken regularly throughout the runs and showed that the rotors were floating under nearly all running conditions, with consequently very light loads on the turbine adjusting blocks.

Auxiliary exhaust connections are arranged at different stages throughout the turbines, so that full use can be made of any available exhaust steam which the feed heater does not use and would otherwise be discharged directly into the condensers.

On the completion of the trials the turbine bearings and thrust rings were opened out for examination and found to be in splendid condition. The turbine casings were not opened out for inspection for the reason that the operation of the turbines throughout the trials was very satisfactory. Further, the joints of the turbine casings never showed any signs of leaking under full-power conditions, and it was therefore considered unnecessary to open up the turbines.

The turbines were built by the New York Shipbuilding Company under license from the Parsons Marine Steam Turbine Company.

### BOILERS.

The boilers are of the Thornycroft Express type, three in number, located in two boiler rooms, one in forward room and two in after room, all on the centerline of the ship, flanked by the coal bunkers and with a cross bunker leading to each boiler for use in action. The total heating surface is 14,493 square feet and the grate surface 270.75 square feet, working

pressure 240 pounds per square inch. The forward boiler is fitted with an auxiliary oil-fuel plant, worked on the mechanical-spray pressure system. The stokehold is closed, and air forced in by means of blowers, one to each boiler.

## PROPELLERS.

There are three propellers, one on each shaft; they are cast solid, of manganese bronze. Each has three blades, machined true to shape. The diameter is 66 inches, pitch 62 inches; projected area of the three blades = 1,716 square inches; developed area = 1,959 square inches. The wing screws turn outboard, the center screw is right-handed; ship going ahead.

The ratio of projected to disc area is .50.

Propellers were carefully balanced.

## AUXILIARY MACHINERY.

The forced-draft outfit consists of three (one for each fireroom) No. 5 double-inlet Troy Sirocco fans driven by Terry steam turbine engines. The capacity of each blower is 30,000 cubic feet at 1,200 revolutions with static pressure of 3 inches of water.

There are two ventilators in each boiler compartment—six in all—and two in the engine room.

The vessel has two evaporators and two distillers of the "Reilly" vertical multicoil navy type. The evaporators have a combined nominal capacity of 9,000 gallons per 24 hours for boiler feed, and of 6,000 gallons per 24 hours for potable water. The combined nominal capacity of the distillers is 6,000 gallons of potable water per 24 hours.

The two main condensers are each cylindrical in form, 5 feet inside diameter by 10 feet 8 inches between tube sheets, with a cooling surface of 4,272 square feet.

An auxiliary condenser of about 400 square feet cooling surface is connected through an auxiliary exhaust pipe to all the auxiliary machinery.

A feed and filter tank of about 650 gallons capacity is provided in the engine room.

The feed-water heater of Schutte-Koerting type, located in the engine room, has a capacity of 100,000 pounds per hour from 90 degrees to 200 degrees F. with 10 pounds back pressure. Exhaust steam is the heating agent. An oil tank for forced lubrication, of about 150 gallons capacity, is fitted in the engine room.

Steam-driven oil pumps are provided for supplying lubricating oil at a pressure of 10 pounds per square inch.

There is a Schutte-Koerting film oil cooler in the engine room, capable of cooling the amount of oil used in the turbine lubrication.

Water service is supplied from existing hose connections on the pumps.

Three steam ash hoists of the navy type are provided, as well as three 4-inch heavy ash ejectors.

Lifting gear is fitted wherever required for the lifting of machinery for overhauling and repairing.

Hand-turning gear is fitted on each shaft line, as is also a torsion meter of the Fottinger type.

PUMPS.

| *No.* | *Name.* | *Size.* | *Type.* |
|---|---|---|---|
| 2 | Main air | 12 × 26 × 18 | Twin, vertical, beam, single-acting; one steam cylinder. |
| 2 | Main circ | 8 × 7 eng., 18" suction and discharge. | Vertical, single engine, centrifugal pump. |
| 2 | Main feed | 13 × 9 × 6 | Simplex, vertical, double-acting, single. |
| 3 | Auxiliary feed | 9 × 6 × 16 | Simplex, vertical, double-acting, single. |
| 1 | Auxiliary air | 6 × 10 × 8 | Vertical single-acting. |
| 1 | Aux. circ | 5 × 5 engine 6" suction and discharge. | Vertical, single engine, centrifugal pump. |
| 2 | Fireroom, fire and bilge. | 10 × 7 × 12 | Single, vertical, double-acting. |
| 2 | Engine room, fire and bilge. | 7 × 7 × 12 | Single, vertical, double-acting. |
| 1 | Evaporator feed | 4½ × 5 × 6 | Simplex, vertical, double-acting, single. |
| 1 | Distiller fresh water. | 3½ × 4 × 4 | Simplex, vertical, double-acting, single. |
| 1 | Oil-cooler circ | 6 × 7 × 8 | Vertical, double-acting, single. |
| 2 | Lubricating oil | 6 × 5½ × 8 | Simplex, vertical, double-acting, single. |
| 2 | Fuel-oil supply | 4½ × 2½ × 6 | Simplex, vertical, double-acting, single. |

With the exception of those of the centrifugal type the pumps are of the Warren Company make. The former were built by the New York Shipbuilding Company.

There is one windlass of Hyde vertical type, with two wildcats suitable for 1⅝-inch stud link chain. The windlass is driven by a 9 × 9 vertical engine, and is capable of hauling in both chains at 6 fathoms per minute.

Two double-headed electric winches, each capable of lifting 2,000 pounds at a speed of 100 feet per minute, are provided.

The steering gear consists of a 10 × 8-inch double horizontal Hyde engine with screw gear, capable of putting rudder from hard-over to hard-over in 25 seconds, when the vessel has a speed of 20 knots an hour.

Two Armstrong, Whitworth air compressors, each compressing 10 cubic feet to 2,500 pounds pressure in 70 minutes, are provided, with piping, valves, etc., for operating the torpedo tubes. Two 10-cubic foot reservoirs and two charging columns are supplied. A pneumatic main is also led from these compressors through the machinery space for use in running pneumatic tools, for blowing soot off boiler tubes, and for operating the fuel-oil service pumps when steam is not available.

There is one 2-ton refrigerating machine of the "Allen" dense-air commercial type, steam driven, connected up for maintaining cold-storage space of about 1,600 cubic feet. This machine has an ice tank, and can make 5 cwt. of ice per day in the tropics, in addition to maintaining the cold storage.

The ship is provided with two 50-kw. turbo-generating sets of General Electric Company type; revolutions about 3,300 per minute, 125 volts pressure at terminals.

Two 24-inch hand-controlled searchlights are fitted. Electric night-signal apparatus is provided. The vessel is equipped with an electric whistle operator. The usual warship outfit of telegraphs and other means of interior communication are provided.

The engineer's workshop is fitted with the following tools, arranged to work by hand and by electric motor drive;

1. One back-geared screw-cutting engine lathe, with swing of 12 inches over ways and to take 46 inches between centers.
2. One column-shaping machine of 15-inch stroke and 15-inch traverse.
3. One double-geared drilling machine with screw feed.

The living quarters below the main deck are ventilated by means of two 4,000-cubic feet navy-standard electric fans. This system extends to magazines and storerooms below the protective deck.

The quarters are heated by steam coils, the plant being arranged to work at a pressure of 50 pounds.

Provision is made for magazine flooding and sprinkling by leads from the firemain.

## GENERAL DESCRIPTION OF THE TRIALS.

These took place in October last, off the Delaware Capes, the Chinese Government being represented by Capt. C. S. Yang and Lieutenant L. Y. Wong, of the Chinese Navy. They consisted of standardization, four hours at full speed; twenty-four hours at 18 knots; gun trials; steering and maneuvering; also tests of auxiliary machinery supplementing those held at dock.

The trials were considerably interfered with by fog and to a lesser extent by wind and sea; the weather that prevailed during the actual endurance runs was normal; whilst at sea the vessel encountered both wind and sea enough to demonstrate that she is a good sea-boat, also that the machinery worked well under adverse weather conditions.

The details of each trial follow:

## STANDARDIZATION.

The details are given in Table No. 1; the displacement, draught, etc., is as follows:

TABLE NO. 1.—STANDARDIZATION RUNS.

Course at Delaware Breakwater. Date, October 5th, 1913.

| Direction | S. | N. | S. | N. | S. | N. | S. | N. | S. | N. | S. | ... | S. | N. | S. | N. | S. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Elapsed time, minutes and seconds | 5-55.1 | 5-07.6 | 5-26.8 | 4-22.2 | 4-14.6 | 4-35.3 | 3-18.0 | 3-39.4 | 3-12.1 | 3-14.3 | 2-55.3 | ..... | 2-44.4 | 3-08.5 | 2-43.2 | 3-03.5 | 2-41.2 |
| Group No. | 1 | | | 2 | | | 3 | | | 4 | | | 5 | | 6 | | |
| Run No. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 |
| Starboard shaft, No. 1, K = .042189. | | | | | | | | | | | | | | | | | |
| Meter | 23 | 24 | 25 | 38 | 36 | 37 | 70 | 73 | 75 | 98 | 93 | ..... | 112 | 111 | 113 | 124 | 124 |
| Revolutions per minute | 263.0 | 270.5 | 273.5 | 339.6 | 333.7 | 332.9 | 450.3 | 458.7 | 461.5 | 526.5 | 508.2 | ..... | 555.1 | 546.1 | 561.8 | 571.2 | 579.6 |
| Shaft horsepower | 255.2 | 273.9 | 288.5 | 544.4 | 506.8 | 519.5 | 1,330.0 | 1,414.0 | 1,460.0 | 2,177.0 | 1,994.0 | ..... | 2,623.0 | 2,557.0 | 2,678.0 | 2,988.0 | 3,032.0 |
| Center shaft, No. 2, K = .042109. | | | | | | | | | | | | | | | | | |
| Meter | 13 | 14 | 24 | 41 | 41 | 39 | 74 | 72 | 72 | 92 | 87 | ..... | 102 | 99 | 107 | 109 | 109 |
| Revolutions per minute | 253.4 | 256.6 | 255.9 | 326.8 | 327.1 | 325.2 | 430.7 | 438.8 | 440.3 | 494.1 | 481.5 | ..... | 518.6 | 508.0 | 523.1 | 527.1 | 534.6 |
| Shaft horsepower | 138.7 | 149.5 | 258.6 | 564.2 | 564.7 | 534.1 | 1,362.0 | 1,330.0 | 1,335.0 | 1,915.0 | 1,764.0 | ..... | 2,228.0 | 2,118.0 | 2,357.0 | 2,419.0 | 2,454.0 |
| Port shaft, No. 3, K = .042674. | | | | | | | | | | | | | | | | | |
| Meter | 23 | 25 | 25 | 40 | 40 | 38 | 72 | 75 | 76 | 97 | 94 | ..... | 116 | 112 | 118 | 128 | 126 |
| Revolutions per minute | 265.2 | 273.1 | 274.7 | 347.1 | 340.8 | 340.1 | 453.3 | 462.8 | 463.7 | 529.0 | 510.3 | ..... | 560.9 | 550.6 | 563.2 | 576.1 | 587.0 |
| Shaft horsepower | 260.3 | 291.4 | 293.1 | 592.4 | 581.7 | 551.5 | 1,393.0 | 1,481.0 | 1,504.0 | 2,190.0 | 2,047.0 | ..... | 2,776.0 | 2,632.0 | 2,836.0 | 3,147.0 | 3,156.0 |
| Total shaft horsepower | 654.2 | 714.8 | 840.2 | 1,701.0 | 1,653.2 | 1,605.1 | 4,085.0 | 4,225.0 | 4,299.0 | 6,282.0 | 5,805.0 | ..... | 7,627.0 | 7,307.0 | 7,871.0 | 8,554.0 | 8,642.0 |
| Average shaft horsepower | 731.0 | 731.0 | 731.0 | 1,653.2 | 1,653.0 | 1,653.0 | 4,209.0 | 4,209.0 | 4,209.0 | 6,044.0 | 6,044.0 | ..... | 7,467.0 | 7,467.0 | 8,405.0 | 8,405.0 | 8,405.0 |
| Average revolutions per minute: | | | | | | | | | | | | | | | | | |
| For run | 260.5 | 265.7 | 268.0 | 337.8 | 333.9 | 332.7 | 446.7 | 453.4 | 455.2 | 516.5 | 500.0 | ..... | 544.9 | 534.9 | 549.3 | 558.1 | 567.1 |
| For group | 265.0 | 265.0 | 265.0 | 334.6 | 334.6 | 334.6 | 452.2 | 452.2 | 452.2 | 508.25 | 508.25 | ..... | 539.9 | 539.9 | 558.15 | 558.15 | 558.15 |
| Average speed: | | | | | | | | | | | | | | | | | |
| For run | 10.140 | 11.703 | 11.016 | 13.730 | 14.140 | 13.077 | 18.182 | 16.408 | 18.740 | 18.528 | 20.536 | ..... | 21.898 | 19.098 | 22.059 | 19.618 | 22.332 |
| For group | 11.14 | 11.14 | 11.14 | 13.772 | 13.772 | 13.772 | 17.434 | 17.434 | 17.434 | 19.532 | 19.532 | ..... | 20.498 | 20.498 | 20.907 | 20.907 | 20.907 |
| Average slip: | | | | | | | | | | | | | | | | | |
| For run | 23.6 | 13.5 | 19.3 | 20.2 | 16.9 | 22.8 | 20.16 | 28.99 | 19.21 | 29.5 | 19.4 | ..... | 21.08 | 29.89 | 21.15 | 30.95 | 22.63 |
| For group | 17.47 | 17.47 | 17.47 | 19.2 | 19.2 | 19.2 | 24.34 | 24.34 | 24.34 | 24.5 | 24.5 | ..... | 25.5 | 25.5 | 26.4 | 26.4 | 26.4 |
| Pressure by gages: | | | | | | | | | | | | | | | | | |
| Main steam | 220 | 219 | 228 | 210 | 205 | 230 | 215 | 225 | 225 | 215 | 215 | ..... | 235 | 215 | 230 | 230 | 228 |
| High-pressure, belt | 25 | 28 | 28 | 62½ | 62½ | 60 | 135 | 135 | 136 | 173 | 172 | ..... | 173 | 165 | 172 | 171 | 171 |
| High-pressure gage, 2d expansion | 13 | 14 | 14 | 35 | 35 | 30 | 75 | 75 | 75 | 110 | 96 | ..... | 135 | 130 | 142 | 145 | 155 |
| Low-pressure, port, pounds | 14 | 14 | 13 | 2 | 2 | 4 | 8 | 9 | 9 | 17 | 14 | ..... | 22 | 21 | 25 | 26 | 27 |
| Low-pressure, starboard, pounds | 14 | 17 | 17 | 8 | 7 | 6 | 9 | 8 | 8 | 18 | 16 | ..... | 23 | 22 | 26 | 28 | 28 |
| Astern, port, pounds | 30 | 30 | 30 | 30 | 30 | 30 | 30 | 30 | 30 | 30 | 30 | ..... | 30 | 30 | 30 | 30 | 30 |
| Astern, starboard, pounds | 30 | 30 | 30 | 30 | 30 | 30 | 30 | 30 | 30 | 30 | 30 | ..... | 30 | 30 | 30 | 30 | 30 |
| Vacuum, port, inches | 28 | 28 | 28½ | 28 | 28 | 28 | 28 | 28 | 27.8 | 27½ | 27½ | ..... | 27½ | 27½ | 27.8 | 28.8 | 27½ |
| Vaccum, starboard, inches | 28 | 28 | 28 | 28 | 27½ | 28 | 28 | 28 | 27.8 | 27½ | 28 | ..... | 27½ | 28.8 | 27½ | 27½ | 28.8 |
| 4th exp., low-pressure, port, pounds, | 20 | 19 | 22 | 18 | 18 | 19 | 12 | 10 | 10 | 1 | 3 | ..... | 2 | 2 | 2 | 3 | 3 |
| 4th exp., low-pressure, starboard, lbs., | 21 | 20 | 20 | 18 | 20 | 20 | 12 | 10 | 10 | 0 | 2 | ..... | 4 | 3 | 5 | 6 | 7 |
| Gland, steam, pounds | 5 | 6½ | 7½ | 7 | 6 | 6 | 8 | 7½ | 7 | 6½ | 6 | ..... | 6½ | 6 | 8 | 6 | 8 |
| Auxiliary, steam, pounds | 222 | 228 | 335 | 227 | 228 | 235 | 235 | 235 | 230 | 220 | 220 | ..... | 239 | 225 | 240 | 240 | 235 |
| Aux. exhaust heater shell, pounds | 5 | 4 | 3 | 0.8 | 0.8 | 7 | 2.5 | 4 | 3 | 7 | 4 | ..... | 5 | 6 | 4½ | 5.8 | 6 |
| Oil system, pounds | 10 | 11½ | 11½ | 9½ | 10½ | 11 | 11 | 10½ | 11½ | 11½ | 11 | ..... | 11½ | 11½ | 11½ | 11½ | 11½ |
| Distribution, aux. exhaust | | | | | | | Fresh | water | heat | and | 4-low | pressure. | | | | | |

Run No. 12.—This run was discarded.

| | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Barometer, inches | 29.9 | 29.9 | 29.9 | 29.9 | 29.9 | 29.9 | 29.9 | 29.9 | 29.9 | 29.9 | 29.9 | ..... | 29.9 | 29.9 | 29.9 | 29.9 | 29.9 |
| Temperatures, degrees Fahrenheit : | | | | | | | | | | | | | | | | | |
| Engine room | 110 | 110 | 106 | 106 | 108 | 108 | 110 | 110 | 106 | 104 | 104 | ..... | 104 | 102 | 102 | 102 | 104 |
| Feed water | 160 | 161 | 145 | 131 | 120 | 160 | 133 | 134 | 132 | 129 | 131 | ..... | 130 | 135 | 132 | 131 | 128 |
| Oil to cooler | 99 | 100.5 | 102 | 105 | 107 | 107½ | 110 | 112 | 107 | 108 | 109 | ..... | 109 | 109½ | 109½ | 109 | 109½ |
| Oil from cooler | 91 | 94 | 97 | 97 | 101½ | 102 | 104 | 105 | 103 | 103 | 103 | ..... | 104 | 104 | 103 | 103 | 104 |
| Water to cooler | 62½ | 62½ | 62½ | 62½ | 62½ | 62½ | 62½ | 62½ | 62½ | 62½ | 62½ | ..... | 62½ | 62½ | 62½ | 62½ | 62½ |
| Water from cooler | 68 | 68 | 68 | 68 | 68 | 68 | 68½ | 68½ | 69 | 68½ | 69½ | ..... | 69 | 69 | 69 | 68½ | 69 |
| Main injection, port | 62½ | 62½ | 62½ | 62½ | 62½ | 62½ | 62½ | 62½ | 62½ | 62½ | 62½ | ..... | 62½ | 62½ | 62½ | 62½ | 62½ |
| Main injection, starboard | 62½ | 62½ | 62½ | 62½ | 62½ | 62½ | 62½ | 62½ | 62½ | 62½ | 62½ | ..... | 62½ | 62½ | 62½ | 62½ | 62½ |
| Overboard, port | 86 | 85 | 86 | 87 | 88 | 88 | 91 | 91 | 90 | 93 | 95 | ..... | 93 | 92 | 91 | 91 | 93 |
| Overboard, starboard | 86 | 86 | 86 | 87 | 88 | 88 | 92 | 91 | 90 | 94 | 95 | ..... | 95 | 95 | 95 | 94 | 95 |
| Hot well | 76 | 76 | 76 | 77 | 77 | 77 | 84 | 83 | 84 | 88 | 88 | ..... | 89 | 89 | 89 | 92 | 92 |
| Feed tank | 93 | 94 | 93 | 93 | 92 | 92 | 92 | 93 | 92 | 93 | 94 | ..... | 93 | 93 | 93 | 94 | 95 |
| R.p.m of auxiliary machinery : | | | | | | | | | | | | | | | | | |
| Main circulating pump, port | 254 | 252 | 252 | 250 | 250 | 232 | 260 | 270 | 270 | 260 | 260 | ..... | 268 | 276 | 280 | 280 | 284 |
| Main circulating pump, starboard | 254 | 250 | 250 | 242 | 250 | 240 | 260 | 282 | 270 | 264 | 260 | ..... | 266 | 270 | 278 | 276 | 264 |
| Main air pump, port | 28 | 28 | 30 | 30 | 30 | 26 | 34 | 32 | 20 | 40 | 36 | ..... | 36 | ..... | 30 | 30 | ..... |
| Main air pump, starboard | 28 | 28 | 30 | 30 | 31 | 32 | 32 | 34 | 30 | 24 | 26 | ..... | 23 | 30 | 30 | 24 | ..... |
| Main feed pump, port | 13 | 13 | 13 | 14 | 17 | 10 | 25 | 24 | 18 | 24 | 24 | ..... | 30 | 20 | 24 | 24 | 24 |
| Main feed pump, starboard | 3 | 3 | 3 | 4 | 4 | 4 | 4 | 4 | 10 | 12 | 7 | ..... | 14 | 16 | 15 | 16 | 16 |
| Lubricating oil pump, port | 28 | 32 | 36 | 36 | 36 | 36 | 44 | 46 | 70 | 70 | 72 | ..... | 72 | 68 | 68 | 72 | 68 |
| Fireroom No. 1—Boiler No. 1 : | | | | | | | | | | | | | | | | | |
| Boiler steam | 222 | 225 | 230 | 230 | 210 | 230 | 222 | 231 | 230 | 222 | 222 | ..... | 240 | 225 | 240 | 240 | 240 |
| Water level in glass | 2-G | 2-G | 1¼-G | ½-G | 2-G | ½-G | 2-G | 2-G | 1¼-G | 2-G | 2-G | ..... | 2½-G | 2½-G | 2-G | 2-G | 2-G |
| Air pressure, inches water | .7 | .7 | .7 | .5 | .5 | .5 | 1 5 | 2.3 | 2 3 | 4.2 | 4.2 | ..... | 4.3 | 4.3 | 4.5 | 4 5 | 4 5 |
| R.p.m. fan No 1 | 500 | 500 | 500 | 425 | 425 | 425 | 750 | 900 | 900 | 1,200 | 1,200 | .... | 1,200 | 1,200 | 1,240 | 1,240 | 1,240 |
| Thickness fires, inches | 8 | 8 | 8 | 8 | 8 | 8 | 10 | 10 | 10 | 10 | 10 | ..... | 10 | 10 | 10 | 10 | 10 |
| Fireroom temperature, degrees F | 104 | 104 | 104 | 101 | 101 | 101 | 102 | 102 | 102 | 90 | 90 | ..... | 90 | 88 | 90 | 90 | 90 |
| Fireroom No. 2—Boiler No. 2 : | | | | | | | | | | | | | | | | | |
| Boiler steam | 222 | 221 | 232 | 225 | 211 | 232 | 223 | 232 | 231 | 227 | 222 | ..... | 240 | 229 | 240 | 240 | 240 |
| Water level in glass | 2-G | 2½-G | 2½-G | 2½-G | 3-G | 2½-G | 2½-G | 2-G | 2½-G | 2½-G | 1½-G | ..... | 2½-G | 2½-G | 2-G | 2-G | 2½-G |
| Air pressure, inches water | .7 | .7 | .7 | .6 | .7 | .5 | 1.5 | 2.1 | 2.1 | 4.3 | 4 3 | ..... | 4.3 | 4.3 | 4.3 | 4.3 | 4.3 |
| R.p.m of fan No. 2 | 500 | 500 | 500 | 475 | 500 | 450 | 750 | 850 | 850 | 1,220 | 1,200 | ..... | 1,200 | 1,200 | 1,220 | 1,220 | 1,220 |
| Fireroom temperature, degrees F | 95 | 95 | 100 | 100 | 100 | 112 | 108 | 98 | 100 | 102 | 102 | ..... | 104 | 102 | 102 | 102 | 102 |
| Thickness fires, inches | 8 | 8 | 8 | 8 | 8 | 8 | 8 | 8 | 8 | 10 | 10 | ..... | 10 | 12 | 12 | 12 | 12 |
| Fireroom No. 2—Boiler No. 3 : | | | | | | | | | | | | | | | | | |
| Boiler steam | 220 | 222 | 234 | 222 | 214 | 229 | 219 | 230 | 232 | 225 | 223 | ..... | 240 | 226 | 240 | 240 | 240 |
| Water level in glass | 2-G | 2-G | 2-G | 2-G | 2-G | 2-G | 2-G | 2-G | 1¼-G | 2-G | 2-G | ..... | 2-G | 2-G | 2-G | 2-G | 2-G |
| Air pressure, inches water | .7 | .7 | .7 | .6 | .7 | .5 | 1.5 | 2.1 | 2.1 | 4.3 | 4.3 | ..... | 4.3 | 4.3 | 4.3 | 4.3 | 4.3 |
| R.p.m. of Fan No. 3 | 500 | 500 | 500 | 475 | 500 | 450 | 750 | 850 | 850 | 1,200 | 1,200 | ..... | 1,200 | 1,200 | 1,220 | 1,220 | 1,220 |
| Thickness fires, inches | 8 | 8 | 8 | 8 | 8 | 8 | 10 | 10 | 10 | 10 | 10 | ..... | 12 | 12 | 12 | 12 | 12 |
| Stack and deck : | | | | | | | | | | | | | | | | | |
| Stack No. 1, temperature, °F | 345 | 340 | 340 | 360 | 340 | 340 | 400 | 440 | 425 | 460 | 440 | ..... | 550 | 585 | 560 | 520 | ..... |
| Stack No. 2, forward, temp., °F | 440 | 465 | 481 | 520 | 540 | 525 | 610 | 620 | 620 | 760 | ..... | ..... | 1,000 | 800 | 900 | 1,060 | ..... |
| Stack No. 2, aft, temperature, °F | 340 | 345 | 340 | 380 | 370 | 380 | 420 | 420 | 425 | 480 | 460 | ..... | 600 | 630 | 760 | 740 | ..... |
| Temperature on deck, °F | 68 | 70 | 72 | 75 | 72 | 76 | 68 | 75 | 68 | 74 | 68 | ..... | 72 | 70 | 74 | 68 | ..... |

Mean draught during high runs, feet and inches.... 14–0¼
Trim by stern during high runs, inches..... 5
Displacement during high runs, tons..... 2,612

The times on the mile were checked; the ranges were clearly visible; moderate swell; wind, moderate to fresh, easterly.

Reduced to a uniform displacement of 2,600 tons, the curve of speed and power is as follows:

| *Speed in Knots.* | *S.H.P.* |
|---|---|
| 12 | 945 |
| 14 | 1,705 |
| 16 | 2,905 |
| 18 | 4,630 |
| 20 | 6,560 |
| 21 | 8,640 |

The torsionmeters used were of Fottinger type.

ENDURANCE RUNS.

These were two in number, four hours at full speed and twenty-four hours at 18 knots.

The accomplishment of these runs was interfered with by fog and wind. On the 6th October, the full-speed run was made; the 18-knot run was commenced on the 8th, abandoned for fog, and not completed until the 12th October, the results are consistent because weather, etc., was the same for the two portions, observations of the consumptions were kept up over periods long enough to establish results.

The quality of the coal left something to be desired; with better coal, results would compare more favorably with those obtained under similar boilers in other ships.

Throughout the actual runs the sea was moderate and weather conditions normal.

For summary of results, see Table No. 2.

## MANEUVERING TRIALS.

These were held in the Delaware Bay off the Breakwater, at the conclusion of the other trials. With the vessel steaming at 20 knots initially, a complete figure eight was made with the helm hard over, each circle was made in a little over three minutes at a diameter of about 300 yards; angles of heel were also noted. Hand-steering gear was tried later at reduced speed.

At 20 knots ahead the telegraphs were rung "full speed astern;" the engines were going astern in 1¼ minutes at 350 r.p.m. The engines were run astern for several minutes, and reversed to full ahead in one minute. At other times during the trials the turbines maneuvered satisfactorily.

TABLE NO. 2.

*Summary of Results on the Endurance Runs.*

| *Item.* | Four hours full speed | Twenty-four hours at 18 knots. |
|---|---|---|
| Mean trial displacement | 2,598 | 2,570 |
| speed, in knots | 20.3 | 18.25 |
| revolutions per minute | 535.2 | 474.3 |
| shaft horsepower | 7,492 | 4,899 |
| Main steam pressure | 224.4 | 221 |
| H.P. belt pressure | 169 | 151 |
| gage, 2d expansion | 127 | 77 |
| L.P. port | 20.2 | 19.6 |
| starboard | 22.2 | 12 |
| Astern turbine, inches | 30 | 28 |
| Vacuum, port, inches | 27.97 | 28 |
| starboard, inches | 27.75 | 28 |
| Auxiliary steam, pounds | 226 | 228 |
| Oil system, pounds | 11.7 | 11.9 |
| Barometer, inches | 30.05 | 30.15 |
| Feed-water temperature, degrees F | 151.0 | 159.0 |
| Oil to cooler, temperature, degrees F | 116.8 | 116.0 |
| from cooler, temperature, degrees F | 107.4 | 107.0 |
| Water to cooler, temperature, degrees F | 61.0 | 65.0 |
| from cooler, temperature, degrees F | 67.4 | 70.4 |
| Main injections, temperature, degrees F | 61.0 | 65.0 |
| Overboard discharge, port, degrees F | 92.4 | 92.1 |
| starboard, degrees F | 95.0 | 92.3 |
| Hotwell, degrees F | 90.5 | 87.0 |
| Feed tank, degrees F | 96.6 | 90.3 |

| *Fireroom No. 1.* | Four hours full speed. | Twenty-four hours at 18 knots. |
|---|---|---|
| Boiler, steam | 235 | 230 |
| Air pressure, inches | 4.1 (water) | 1.8 |
| Fans, r.p.m | 1,197 | 768 |
| Temperature, degrees F | 83.0 | 102.0 |
| *Fireroom No. 2.* | | |
| Boiler, steam | 233 | 232 |
| Air pressure, inches | 4.1 | 1.4 |
| Fans, r.p.m | 1,177 | 728 |
| Temperature, degrees F | 87.3 | 101.0 |
| Stack temperatures, degrees F | 465 | 440 |
| Deck temperatures, degrees F | 66.3 | 72.5 |
| Engine-room, temperature, degrees F | 102.0 | 106.0 |
| Coal, tons per hour | 7.66 | 5.36 |
| pounds per S.H.P. hour | 2.29 | 2.45 |
| Water rate, pounds per S.H.P. per hour all purposes | 15.45 | 17.50 |

# A METHOD OF CLOSING PORES ON HOLLOW BRASSES.

SUBMITTED BY CAPT. F. W. BARTLETT, U. S. N.

On the U. S. S. *Montana* it was found that the six bottom brasses for the main shaft of each engine were porous, so that water from the water service mingled with the oil of the journals to a slight extent. While these leaks were not serious they were considered sufficiently important to be investigated and stopped if possible.

These brasses were, of course, inspected at the place of manufacture and tested to a certain pressure. About a year and a half ago they were rebabbitted and subjected to a pressure of twenty (20) pounds at this navy yard, and were tight. After a trip from this navy yard to Philadelphia, thence to Smyrna, Turkey, and return by approximately the same route, it was found that the brasses were leaking slightly. The babbitt metal of the brasses, however, was in excellent condition and the leaks were not where the babbitt was located, but at other points of the brasses and at points where the babbitt was cut away. The moisture apparently found its way through small crevices to points under the babbitt and thence to the edges of the babbitt, where it would exude slowly from under the babbitt metal. In some cases small drops appeared on the brasses where there was no babbitt. The leak from the worst of the brasses was not over a teaspoonful an hour.

Inasmuch as the babbitt was in good condition, the only method of improving the situation would be to make new brasses, unless some method could be devised for closing the pores of the metal where it leaked. As it is a very expensive proposition to fit new brasses, many methods were considered for making tight the old ones, and finally the method described below was attempted and found successful.

METHOD USED.

The brasses were first filled with gasoline and subjected to a pressure of about twenty (20) pounds and left in this condition for about twelve (12) hours. It was found that the gasoline worked its way through the crevices and appeared at the small openings, and in some cases carried with it what appeared to be particles of red lead, giving the impression that an attempt had been made at some time to close these crevices by forcing in red lead under pressure; but this could not be proved. The idea of using the gasoline was to thoroughly cleanse the crevices of oil and water. The gasoline was then removed and air blown through for four (4) hours and the brasses were then filled with shellac and a pressure of seventy-five (75) pounds of compressed air was put on this shellac and it was kept in this condition for about four (4) hours, the idea being that the shellac would not settle materially in that time. It was found that the alcohol appeared to push ahead of it small particles of water or gasoline, and appeared as drops at the small crevices, and in some cases the shellac could be seen clearly to have passed through these crevices and to have arrived on the outside as a brown liquid. This, however, soon congealed and no further leaks took place. The shellac was then removed and compressed air blown through the brasses for three or four hours with the idea of setting the shellac on the surfaces within, although it was realized that the shellac in the small crevices would be in no way affected. After this the brasses were filled with water and tested to a pressure of seventy-five (75) pounds, with no sign of leakage after an hour of observation.

Shellac was the material which seemed best adapted for this purpose over all others considered, but it is believed that some other material might be found to accomplish this purpose, and a material which would not become soft at as low a temperature as shellac. Experiments along that line should be conducted.

It is believed that the shellac has filled the small crevices completely, and there seems to be no reason why this shellac should ever be disturbed, as the course of the water through

the bearing cannot effect the material in the tiny crevices. In addition, a pressure of seventy-five (75) pounds is much higher than has been customary to use in testing these brasses, and it was considered that twenty (20) pounds was sufficient as the static pressure of the water on board ship with connections as ordinarily made is less than ten (10) pounds.

### PRESSURE ON WATER SERVICE.

Another idea that presented itself in studying this question was why the brasses leaked when they were tight as installed with a pressure of twenty (20) pounds. It appeared, then, that there is always another connection to the water service from a pump, and that this connection had no reducing valve on it as installed in the service, so that the entire pressure (up to the capacity of the pump) might be placed upon this water-service line. This would readily account for the leakage of the brasses where tight as tested up to twenty pounds. As a rule the pump to which the water service is attached is the fire and bilge pump, so that the pressure that might be put upon the water service would be that at which the fire-main relief valve is set. It is believed that it is advisable to have reducing valves on these connections to the water service.

### TESTS OF SHELLAC.

The shellac in its dry form was tested in a sand bath, with results as shown in the following table:

150 degrees F.—Loses elasticity.
175 degrees F.—Sticky; pieces stick together.
250 degrees F.—Like thick molasses.
300 degrees F.—Like thin syrup.
350 degrees F.—Begins to harden.
390 degrees F.—Hard. Like particles of dust adhering.
400 degrees F.—Bakes to bottom of bath.
450 degrees F.—Charred, and lessening in quantity very materially.

From this point up to 700 degrees F. the material simply shrank away, and at 700 degrees was a black dust and very small in quantiy.

It would appear from the above that up to 150 degrees F. the shellac retains practically its original consistency, so that it surely will close the pores until that temperature is reached, and that temperature is such as should never be allowed on main brasses or any other brasses. However, it will be noted that up to 250 degrees this shellac has still the consistency of thick molasses, and it is believed that the water pressure behind this shellac would simply force this liquid still more firmly into the crevices, if possible, and might force out a few drops to join the oil, but it would take a considerable period of high heat for all the shellac to be driven out of these fine crevices.

Of course this is simply a trial, and may not be successful; but is thought worthy of the attention of those interested in engineering.

## METHOD OF GRINDING IN AND DRESSING UP DAMAGED THRUST BEARING.

By Lieut. Commander H. C. Dinger, U. S. N.

In the four-shaft Parsons turbine installation where a H.P. and I.P. cruising turbine are employed, the two cruising turbines being on the inboard shafts, there is considerable excess propeller thrust on the L.P. thrust collars whose shaft also carries the I.P. cruising turbine when going up to speeds near the limit of this combination. Under these conditions when the propeller thrust which is taken on the lower half rings, the L.P. thrusts of the shafts that carry the I.P. cruising turbine are quite susceptible to heating, and this thrust has been burned out on several vessels.

There is usually sufficient surface in the L.P. thrust collars, and the thrust will operate quite satisfactorily provided a sufficient number of the collars bear and their surface is true.

On one vessel this thrust operated without trouble for a long period, when one day when it was desired to speed up to 19 knots quickly the thrust heated rapidly and could not be cooled till the combination was shifted so as to release the excess thrust and allow the rotor to move out.

Previous to this speeds as high as 20 knots had been run on this combination for long periods with thrust quite cool.

In connection with this matter it has been noted that the rotor comes in harder and there is considerable more propeller thrust when in shoal water. The worst condition for heating is therefore in speeding up quickly on the I.P. combination in shoal water. When the vessel is by herself this condition can be avoided to some extent, but when in formation any speed called must be quickly given.

On anchoring the heated thrust was examined, and about

six of the collars found to have been quite hot, and surface of nearly all collars and rings were cut and roughened. The surfaces were cleaned and stoned; but as there were only a few hours available to do this work, a thorough job could not be made of it.

On trying this thrust again on the I.P. combination it again heated up at about 17 knots.

During an overhaul week, a few days later, the thrust was carefully examined.

*Starboard Low-pressure Thrust Bearing.*

Distances in thousandths of an inch between collars on shaft and collars on bearing.

GO-AHEAD SIDE.

| Collar. | I | O | I | O | I | O | I | O |
|---|---|---|---|---|---|---|---|---|
| | 0 degree. | | 90 degrees. | | 180 degrees. | | 270 degrees. | |
| 1 | 54 | 56 | 61 | 53 | 62 | 60 | 56 | 62 |
| 2 | 54 | 56 | 61 | 53 | 62 | 50 | 56 | 62 |
| 3 | 55 | 56 | 62 | 53 | 63 | 60 | 57 | 62 |
| 4 | 57 | 56 | 64 | 53 | 64 | 60 | 58 | 62 |
| 5 | 57 | 57 | 64 | 54 | 64 | 61 | 59 | 63 |
| 6 | 56 | 57 | 63 | 54 | 63 | 61 | 58 | 63 |
| 7 | 56 | 57 | 63 | 55 | 64 | 62 | 58 | 64 |
| 8 | 55 | 57 | 62 | 55 | 62 | 62 | 57 | 64 |
| 9 | 54 | 57 | 61 | 54 | 62 | 62 | 57 | 63 |
| 10 | 55 | 58 | 62 | 55 | 63 | 62 | 57 | 64 |
| 11 | 55 | 57 | 62 | 54 | 62 | 62 | 57 | 63 |
| 12 | 55 | 56 | 61 | 54 | 62 | 61 | 57 | 62 |
| 13 | 55 | 56 | 61 | 55 | 62 | 61 | 57 | 63 |
| 14 | 56 | 57 | 52 | 55 | 63 | 62 | 57 | 63 |
| 15 | 56 | 57 | 62 | 55 | 62 | 62 | 57 | 64 |
| 16 | 55 | 56 | 50 | 54 | 62 | 60 | 57 | 62 |
| 17 | 57 | 58 | 63 | 56 | 64 | 63 | 59 | 64 |
| | | | Backing face. | | | | | |
| 1 | 34 | 38 | 33 | 31 | 37 | 28 | 43 | 34 |
| 2 | 38 | 38 | 36 | 31 | 37 | 28 | 44 | 34 |

The clearance on bottom rings and go-ahead side of collars were measured by feeler gages, with shaft in position, at intervals of 90 degrees. The readings are given in table. The readings conclusively showed that the surfaces of the collars

did not run true. The heated collars also showed that the most intense heat was at one part of the circumference, and this was the same part of the circumference on all of the collars. Squares and straight edges placed on the surfaces of the collars also showed that the collars were far from being true, and that they were not parallel to each other or normal to the axis of the shaft.

Spotting the shaft collars with red lead and bringing the thrust wings up to bear and then revolving shaft also clearly showed that the collars ran out and the collars that bore touched only in spots on one side.

The flange of the thrust shaft was tramed to see whether the shaft ran out. The differences for different points of the circumference were so slight that it was apparent that the shaft itself was not bent. From the measurements taken it appeared that the surfaces of collars had not been machined true, but that there had been sufficient bearing surface to allow the thrust to run cool till, on account of some special circumstances, due perhaps to speeding up quickly, an excessive load was produced and the heating resulted.

To remedy the defective conditions the following was done: The lower thrust-block shell, carrying the rings taking the go-ahead propeller thrust, was carefully cleaned up and put in place. The surface of the collars was carefully covered with red lead, the thrust brought up to touch, and then the shaft revolved. The first spotting showed number 6 collar to bear hard in two spots and about five other collars bearing slightly in streaks for about 40 degrees on one side. The spots were then filed and the spotting process repeated. The second spotting showed about eight collars bearing in streaks part of the way around. The third spotting showed about ten collars bearing in rather heavy streaks about half way around. Filing was then stopped, as it was feared to take off too much metal, and time was also limited. A novel grinding-in process was then tried. The lower thrust block was put in place. A dummy adjustment gig, which permits of rotors being moved in or out with a ratchet gear, was rigged over the thrust collars to keep

rotor in place. A mixture of bath brick and oil was put on thrust bearing and the turbine rotated at about 25 revolutions with steam. The collars were brought to bear by means of the gig and were moved up slightly tighter as the grinding proceeded. The supply of powdered bath brick and oil was continually replenished while the grinding was going on. The brick dust and oil ground the surfaces in contact and tended to smooth them up. As the collars are of hard steel and the thrust rings of composition, most of the grinding will be done on the rings; however, some grinding and smoothing up of steel surfaces will also result. This grinding was continued for about 2½ hours, when it was stopped. The thrust block was taken out and all parts of rings and collars carefully cleaned and then all parts were replaced.

After these adjustments were made this thrust, though regarded with suspicion, ran satisfactorily up to nearly 18 knots on this combination for nearly three months, when, on another occasion, it was necessary to speed up quickly in shoal water, it again heated badly.

The vessel arrived at the navy yard shortly after for a two weeks' stay, and thus an opportunity for more carefully overhauling this thrust presented itself. The shell containing the rings was taken to the shop and the edges of counter sinks and all oil grooves were carefully milled out and the surfaces carefully smoothed up.

The surfaces of collars were carefully dressed up with file and scrapers by two navy-yard mechanics especially expert at this work. Then the process of spotting the collars with red lead and filing down high spots was repeated, but this was done more often than on first attempt. This work took two machinists six days.

When this spotting and filing had progressed so far that a fair bearing was obtained on most of the collars and all of the collars touched at some points, the surfaces were again ground in a bath of oil and brick dust for about three hours, running the turbine with steam at about 30 revolutions per minute. Everything was again carefully cleaned out.

This process of cleaning out, spotting and filing and finally grinding-in, produced fair bearing surfaces and insured surfaces being smooth.

The thrust was tried soon after this to the practical limit of the I.P. combination and operated satisfactorily.

This method of grinding in a bearing is perhaps somewhat novel, but it appears to be quite effective and accomplishes what might otherwise require a large amount of labor.

# REPORT OF METALLOGRAPHIC PHYSICAL TESTS OF BROKEN CRANKSHAFT.

MADE AT U. S. NAVAL EXPERIMENTAL STATION, ANNAPOLIS, MD.

REPORT PREPARED BY D. J. MCADAM.

1. The Bureau of Steam Engineering directed this Station to make metallographic and physical tests of the material of this crankshaft, in order to determine the character of the metal and the cause of the break.

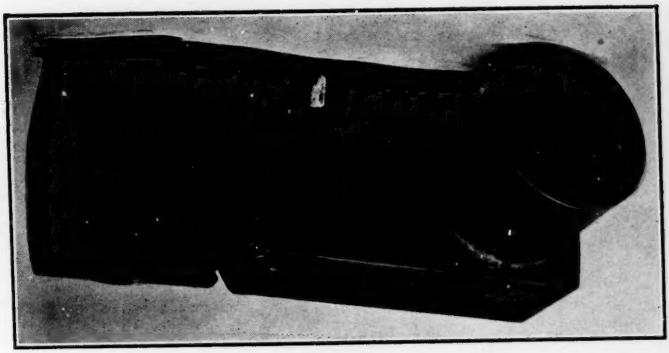
2. The crankshaft had broken through the crank web close to the crank pin, as shown in the photographs, Inclosures (L) and (M). The break extended through a portion of the axial hole in the crank pin. The history of the shaft in service was not given.

3. The *Purpose of the Test* was to make a chemical, physical and metallographic examination of the material of the shaft, and thus determine the cause of the break.

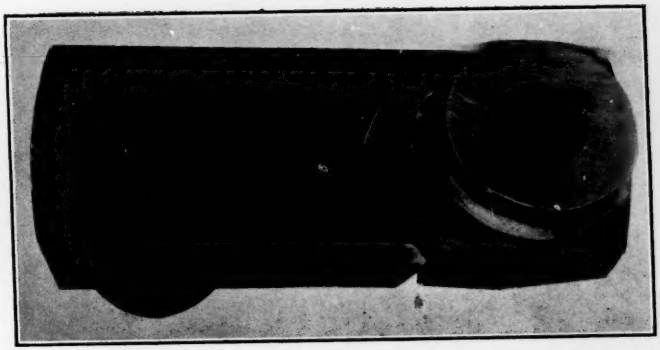
4. *Manner of Test.*—After photographs were taken, the shaft was cut into sections, as shown in the sketch, Inclosure (K), and tensile and metallographic specimens were cut from these sections as shown in the same sketch. Millings were also taken for chemical analysis.

5. *Results.*—(*a*) The results of the chemical anlysis were as follows:

| | Average. | No. of determinations. |
|---|---|---|
| Carbon | 0.346 | 3 |
| Silicon | 0.038 | 2 |
| Sulphur | 0.016 | 3 |
| Phosphorus | 0.015 | 3 |
| Manganese | 0.321 | 2 |
| Copper | 0.036 | 1 |
| Nickel | 0.019 | 3 |



INCLOSURE (L).

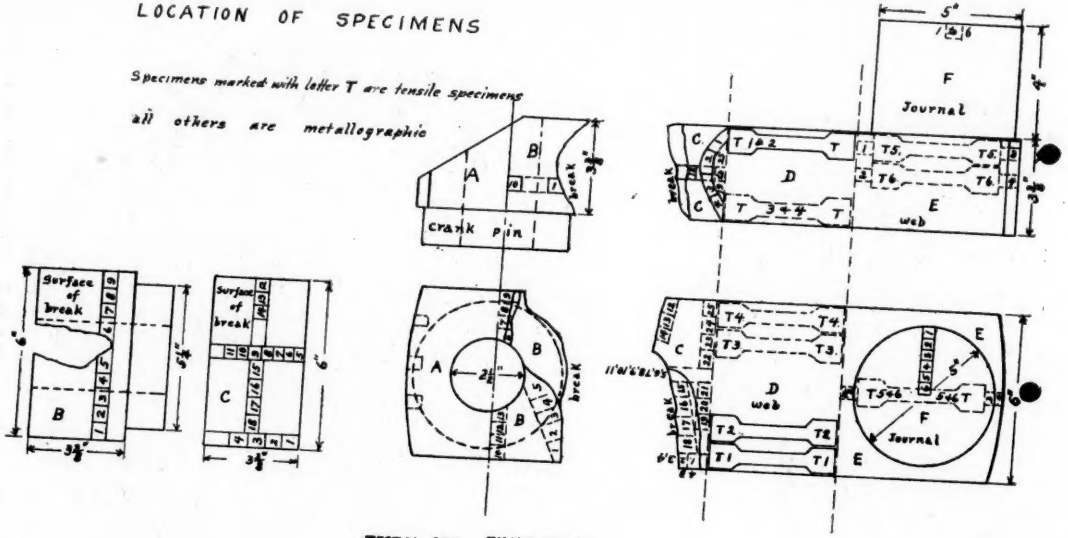


INCLOSURE (M).

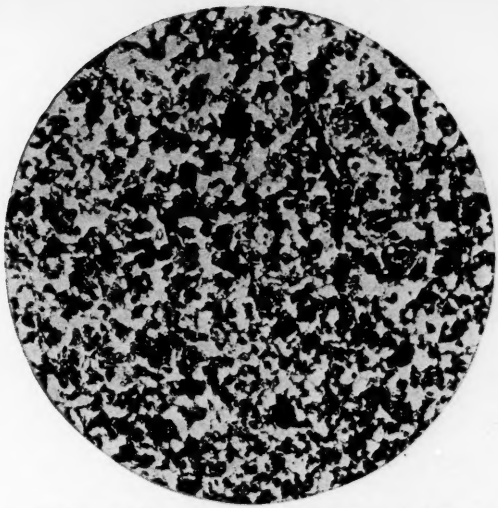
# LOCATION OF SPECIMENS

Specimens marked with letter T are tensile specimens

all others are metallographic




TEST No. 357. ENCLOSURE (K)



CARD NO. 1. ENCLOSURE (A). SPECIMEN C–3, LONGITUDINAL SURFACE. MAGNIFIED 100 DIAMETERS.

*Structure.*—Ferrite, pearlite; granular. White areas are ferrite, dark areas pearlite.
*Remarks.*—This photograph shows a good structure. The grain is small and the ferrite and pearlite are evenly distributed. The specimen was taken from the outer surface of the crank web.

CARD NO. 2. ENCLOSURE (B). MAGNIFIED 100 DIAMETERS. BUTTON TAKEN FROM SPECIMEN B–13, CROSS-SECTIONAL SURFACE.

*Structure.*—Ferrite, pearlite; coarse network, with ferrite in lines of cleavage. White areas are ferrite, dark areas are pearlite.
*Remarks.*—This photograph shows a coarse structure somewhat like that of cast steel. An outline of a coarse-network structure can be seen, one mesh occupying nearly the whole field. Within the mesh the ferrite is massed in lines of cleavage in three directions, making an angle of about 60 degrees with each other. Such a structure is produced by heating to a very high temperature with subsequent undisturbed cooling.

CARD NO. 3. ENCLOSURE (C). MAGNIFIED 45 DIAMETERS. SPECIMEN TAKEN FROM B–5, LONGITUDINAL SURFACE.

*Structure.*—Ferrite, pearlite; network, with ferrite in lines of cleavage. White areas are ferrite, dark areas are pearlite.

*Remarks.*—This shows the same kind of structure that is seen on Card No. 2, Enclosure (B); but magnification is less, so that several meshes of the coarse-network structure can be seen. The ferrite is massed on the network lines and also on the cleavage lines within each mesh, forming a triangular appearance.

CARD NO. 4. ENCLOSURE (D). MAGNIFIED 45 DIAMETERS. BUTTON TAKEN FROM SPECIMEN C–6, LONGITUDINAL SURFACE.

*Structure.*—Ferrite, pearlite; network, with ferrite in lines of cleavage. White areas are ferrite, dark areas are pearlite.

*Remarks.*—This photograph shows an unusually coarse-network structure with the ferrite on the boundaries of each mesh and also on cleavage lines within the mesh. The lines of cleavage are in three directions; making an angle of about 60 degrees with each other. Such structure is typical of cast steel or steel that has cooled slowly from a temperature near the melting point. The effect of forging has not reached this portion of the metal. The magnification was lower than usual on account of the very coarse structure.

CARD NO. 5. ENCLOSURE (E). MAGNIFIED 45 DIAMETERS. BUTTON TAKEN FROM SPECIMEN C–6, LONGITUDINAL SURFACE.

*Structure.*—Ferrite, pearlite; very coarse network, with ferrite—lines of cleavage. White areas are ferrite, dark areas are pearlite.

*Remarks.*—This photograph shows another portion of the same specimen seen on Card No. 4, Enclosure (D). Several meshes of an unusually coarse-network structure are partly visible. Within these meshes a triangular structure is seen. The ferrite is massed along the network and cleavage lines. Such a structure is caused by slow cooling from a very high temperature, it has been practically unaffected by forging. The magnification was lower than usual on account of the coarseness of the structure.

CARD NO. 6. ENCLOSURE (F). MAGNIFIED 45 DIAMETERS. BUTTON TAKEN FROM SPECIMEN C–22, LONGITUDINAL SURFACE.

*Structure.*—Ferrite, pearlite; remnants of network, with ferrite in lines of cleavage. White areas are ferrite, dark areas are pearlite.

*Remarks.*—This photograph shows a view of the material a few inches away from the break, but near the center of the crank web. By comparison with Cards 4 and 5, it will be seen that the structure here shown is somewhat finer, but it still shows traces of the network and triangular structure, particularly on the right of the picture. The magnification is only 45, so the structure is still very coarse.

CARD NO. 7. ENCLOSURE (G). MAGNIFIED 100 DIAMETERS. BUTTON TAKEN FROM SPECIMEN E–2, CROSS-SECTIONAL VIEW.

*Structure.*—Ferrite, pearlite; granular, with remnants of coarse network. White areas are ferrite, dark areas are pearlite.

*Remarks.*—This photograph shows the structure near the center of the crank web, but distant from the break. Remnants of the network and triangular structure are visible, and the structure is very coarse, but not so coarse as at the break. The broad ferrite grains are lines of weakness.

CARD NO. 8. ENCLOSURE (H). SPECIMEN TAKEN FROM C–20, HEAT TREATED.

*Structure.*—Ferrite, pearlite; granular. White areas are ferrite, dark areas are pearlite.

*Remarks.*—This specimen was heat treated at 788 degrees C. (1,450 degrees F.) for one hour and cooled in ashes. The original structure was about like that shown on Card No. 2, Enclosure (B). The heat treatment has removed the network and triangular structure almost completely, and left a fairly good structure. On account of the original coarseness a larger heat treatment would be necessary to make the material entirely uniform.

(*b*) Six pieces for physical tests were cut from sections D and E as shown in the sketch, Inclosure (K). The results of the tensile tests were as follows:

| Piece No. | 1 | 2 | 3 | 4 | 5 | 6 | Ave. |
|---|---|---|---|---|---|---|---|
| Diameter: | | | | | | | |
| Test piece, inches | .5036 | .5037 | .5038 | .5045 | .5056 | .5049 | |
| Fracture, inches | .3383 | .3394 | .3396 | .3475 | .3448 | .3480 | |
| Area: | | | | | | | |
| Test piece, sq. inches | .1992 | .1993 | .1994 | .1999 | .2008 | .2002 | |
| Fracture, sq. inches | .08989 | .09047 | .09058 | .09484 | .09337 | .09511 | |
| Reduction, per cent | 54.87 | 54.61 | 54.57 | 52.56 | 53.50 | 52.49 | 53.77 |
| Length between punch marks, inches | 2.000 | 2.000 | 2.000 | 2.000 | 2.000 | 2.000 | |
| Elongation: | | | | | | | |
| Inches | .6094 | .6172 | .6797 | .6094 | .6250 | .6094 | |
| Per cent | 30.47 | 30.86 | 33.98 | 30.47 | 31.25 | 30.47 | 31.25 |
| Elastic limit: | | | | | | | |
| Pounds | 7,800 | 6,300 | 6,800 | 7,500 | 7,400 | 7,700 | |
| Pounds per square inch | 39,160 | 31,610 | 34,120 | 37,520 | 36,850 | 38,460 | 36,290 |
| Maximum stress: | | | | | | | |
| Pounds | 14,000 | 13,200 | 13,700 | 14,550 | 13,750 | 13,900 | |
| Pounds per square inch | 70,280 | 66,230 | 68,700 | 72,790 | 68,480 | 69,460 | 69,320 |
| Remarks; fractures, almost silky | dull, fine. | dull, fine. | dull, fine. | dull, fine. | dull, fine. | dull, fine. | |

NOTE.—Class A material is called for in the specifications (Machinery Specifications No. 101, *Paducah*. The requirements for class A forgings are:—Minimum tensile strength, 80,000 pounds; minimum elastic limit, 50,000 pounds; minimum percentage of elongation, 25.

(*c*) Specimens for the metallographic examination were taken from various parts of the shaft, as shown in the sketch, Inclosure (K). About fifty specimens were examined, and eight of these, including one heat-treated specimen, were photographed and are shown on cards 1 to 8, Inclosures (A) to (H). In all of these photomicrographs, the white areas represent ferrite and the dark areas represent pearlite. On account of the coarseness of the structure in some specimens it was necessary to use a lower magnification than usual in some of the photomicrographs.

*Card No. 1*, Inclosure (A), shows specimen C–3, longitudinal surface. The specimen was on the line of the break, but near the edge of the crank web. The structure is good,

the grain size being small and the two constituents being uniformly distributed.

*Card No. 2,* Inclosure (B), shows specimen B–13 on a transverse surface. This material was about two inches from the break, and near the center line of the crank web at its junction with the crank pin. The photograph shows a coarse network structure, one mesh occupying nearly the whole field of view. Within this mesh the ferrite occurs in cleavage lines, making a triangular structure.

*Card No. 3,* Inclosure (C), shows specimen B–5 on a longitudinal surface. The specimen was located on the line of break and near the center line of the crank web. It shows the same kind of structure that is seen on Card No. 2; but the magnification is less, so that several meshes of the coarse network structure are visible. Vague outlines of these meshes can be seen; and within each mesh the ferrite is massed in the cleavage planes, which extend in three directions making angles of 60 degrees with each other.

*Card No. 4,* Inclosure (D), shows specimen C–6. The photograph shows a longitudinal section on the line of break and near the center line of the crank web. An extremely coarse network structure is visible, with the ferrite massed along the cleavage lines within each mesh, as seen on cards 2 and 3. In spite of the low magnification, one mesh occupies a large part of the photograph.

*Card No. 5,* Inclosure (E), shows another portion of the same specimen that is seen on Card No. 4. The magnification is only 45, but portions of only three or four meshes of the network structure are visible. The triangular structure characteristic of cast steel is quite prominent.

*Card No. 6,* Inclosure (F), shows specimen C–22 on a longitudinal surface. The specimen was from a point about two inches away from the line of break and near the center line of the crank web. The structure is not so coarse as that shown on Cards 4 and 5; but it still shows remnants of the network and triangular structure. Taking into account the two different magnifications, 45 and 100, it is evident that the

coarseness of structure is about the same in this specimen as in the specimens shown on Cards 2, 3 and 7.

*Card No. 7,* Inclosure (G), shows specimen E–2. This is a transverse view of a specimen which was distant from the break, but near the center line of the crank web. Remnants of the network and triangular structure are visible, and it compares in coarseness with that seen on Cards 2, 3 and 6.

*Card No. 8,* Inclosure (H), shows specimen C–20 after heat treatment. It was heated at 788 degrees C. (1,450 degrees F.) for an hour and cooled in ashes. The original structure was about like that shown on Card No. 2, Inclosure (B). The heat treatment has practically removed the network and triangular structure, and produced a rather fine-grained material. On account of the coarseness of the original material, a longer heat treatment would be necessary to make it entirely uniform.

These photomicrographs show structures that are typical of those seen on all specimens examined. Near the edges of the crank web the material is fine grained, and the grain size gradually increases as the center line of the web is approached. Near the center line a coarse structure was invariably found, the coarsest structure being near the surface of the break. The specimens from sections A and F, as shown in the sketch, Inclosure (K), showed a fine-grained structure similar to that seen on Card No. 1, Inclosure (A). A number of small spots of slag and manganese sulphide were seen in the various specimens, but their effect on the properties of the steel is negligible.

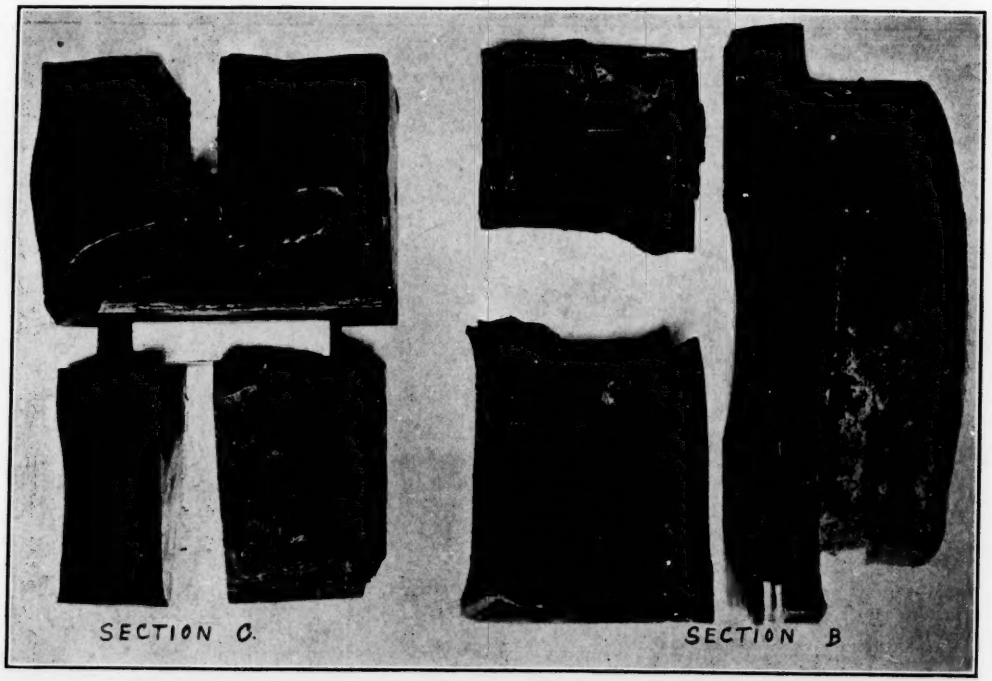
(*d*) *General Appearance of the Shaft Metal.*—On the surface of the break a number of parallel ridges and laminations were visible. When the material was cut by the saw, laminations were visible on the smooth surface also; they apparently consisted of parallel lines of harder and softer metal alternating. These laminations were found in the material near the break and at the junction of the crank web and crank pin. The laminations on the surface of the break are shown in the photograph, Inclosure (I), sections B and C being the same that

are shown in the sketch, Inclosure (K). In the photograph, Inclosure (J), two of the pieces have been turned so as to show the laminations on the cut surfaces. The break occurred along the lines of these laminations.

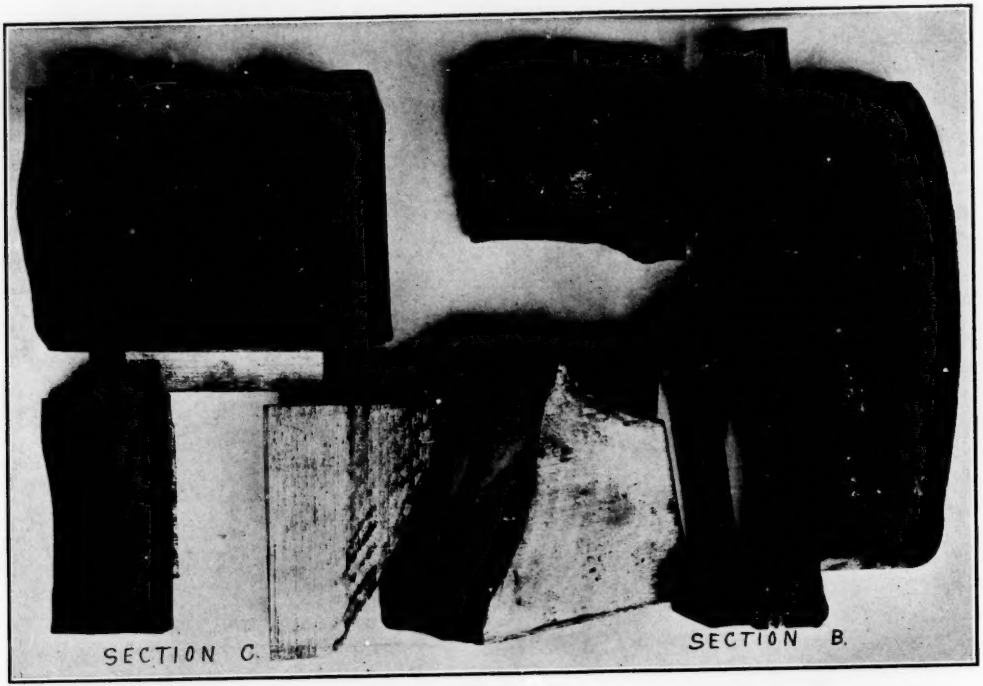
6. *Discussion.*—(*e*) The results of the chemical analysis show that this is a medium carbon-steel of good chemical composition. The percentages of sulphur and phosphorus are well within the required limits.

(*f*) The results of the tensile tests show that this material does not fulfill the requirements of Class A forgings according to the specifications (Machinery Specifications, No. 101, *Paducah*). For Class A forgings the minimum values for tensile strength and elastic limit are 80,000 pounds and 50,000 pounds respectively. In the six tensile specimens tested here, the values for tensile strength and elastic limit fell considerably below these requirements, the average results being 69,320 and 36,290 respectively. This material therefore shows the physical properties of Class B forgings rather than Class A.

(*g*) The results of the metallographic examination show that this material had an unusually coarse structure near the break, and that the structure all along the center line of the crank web was very coarse. Near the edges of the crank web the structure was good. The causes of this structure of the shaft are twofold—improper forging and improper heat treatment. The material near the center of the crank web, and especially near the crank pin, was practically unaffected by the forging process, or by subsequent heat treatment. The structure resembles that of cast steel; it appears as if the material in these portions of the shaft had cooled from near the melting point without any disturbance due to the forging process. Any annealing that the shaft received after forging has affected only the outside of the shaft; it has not been sufficient to remove the coarse structure of the interior. Such a defective structure could have been caused by finishing the forging at a very high temperature, or by using a hammer that was too light for the effect of the blows to reach the interior of the forging. Or the original size of the ingot




INCLOSURE (I).




INCLOSURE (J).

may have been too small in comparison with the size of the forging, so that the material did not receive sufficient working. Bradley Stoughton, on page 173 of his book on "The Metallurgy of Iron and Steel," says: "Once crystals have formed, they cannot be reduced in size except by annealing or by breaking them up by mechanical crushing. These facts are important, because large crystals do not adhere to each other firmly and thus they cause a weak and brittle mass."

Harbord and Hall, on page 841 of their book on "The Metallurgy of Steel," discuss "Unexpected fracture of a steel shaft, which will usually be found to have been made from an ingot of insufficient size or from a hammer insufficiently heavy to insure that the center of the piece has been properly worked. As a case in point, we have the breaking of the shaft of the U. S. despatch boat *Dolphin*. The shaft, 16 inches in diameter, had been made from a 30-inch steel ingot under a ten-ton hammer. The outer surface only had been properly worked, the center remaining in its original raw condition because the hammer did not possess sufficient power to break up the large crystals for more than a short distance from the surface."

(*h*) The laminations, described in paragraph 5 (*d*) above, indicate that the material had been forged by hammer and had not been worked sufficiently to render it uniform. These laminations were planes of weakness along which the break occurred.

7. *Conclusions.*—The chemical composition of this material is satisfactory. The physical tests showed that the material had the physical properties of Class B forgings rather than Class A forgings. The breaking of this shaft was due to brittleness and weakness caused by an unusually coarse-grained structure. The defective quality of the material was caused by ineffective forging followed by insufficient annealing.

# DAMAGE TO U. S. NAVAL AUXILIARY *JASON* AS A RESULT OF EXPLOSION OF DYNAMITE ON STEAMER *ALUM CHINE*.

By Lieutenant R. L. Irvine, Member.

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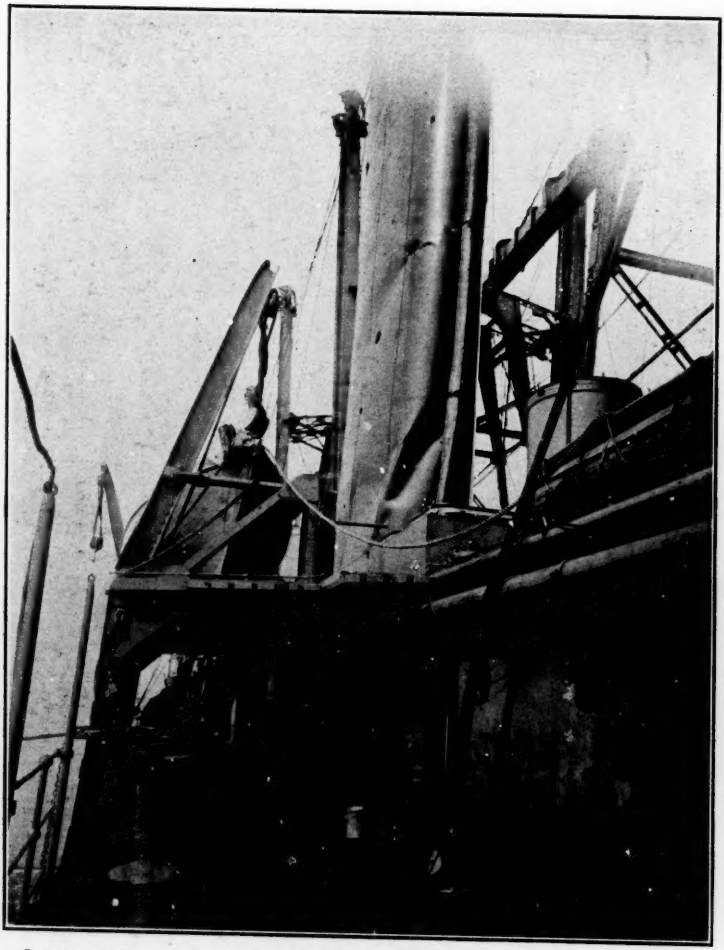
At about 10:35 a. m., March 7, 1913, an explosion of a cargo of dynamite took place on board the steamer *Alum Chine* in the outer harbor at Baltimore, Maryland. The vessel was taking the dynamite on board at the time from two freight cars on a lighter moored alongside. The *Jason* was still in the hands of the contractors, the Maryland Steel Company, Sparrow's Point, Maryland, had just been docked, filled with a full cargo of coal and provisions, had a full crew on board, and was lying at anchor at a distance estimated by various persons to be from 500 to 800 yards distant from the *Alum Chine*. The *Jason* was to depart the following morning for the Delaware Breakwater to conduct her official preliminary acceptance trials.

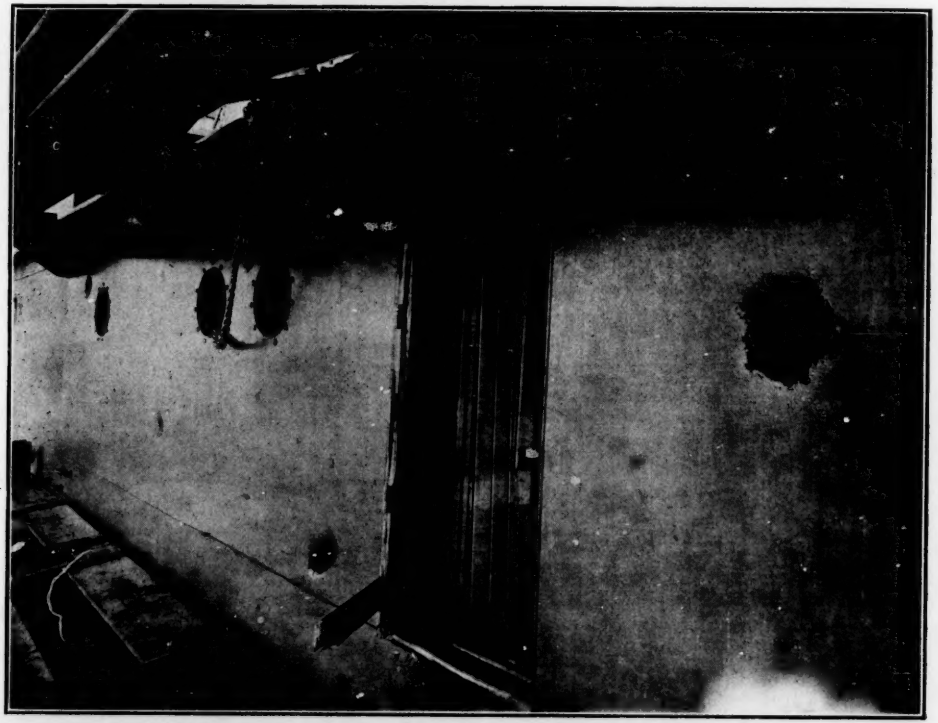
The *Jason* was lying with her port side toward the *Alum Chine*. It is thought that about 300 tons of dynamite exploded. The exact cause of the accident is not known, but prior to the explosion persons on and near the *Jason* noticed smoke issuing from the *Alum Chine*, and workmen running for safety on board. Most of them managed to reach a launch that was secured near the bow, and would have been saved had not the launch returned to the vessel after once having started away, in order to rescue two remaining men. This launch was completely wrecked and sank, and most of the people on board were killed. The *Alum Chine* appears to have been blown into very small pieces.

The accompanying photographs, which were taken immediately after the accident and before anything was cleared up,

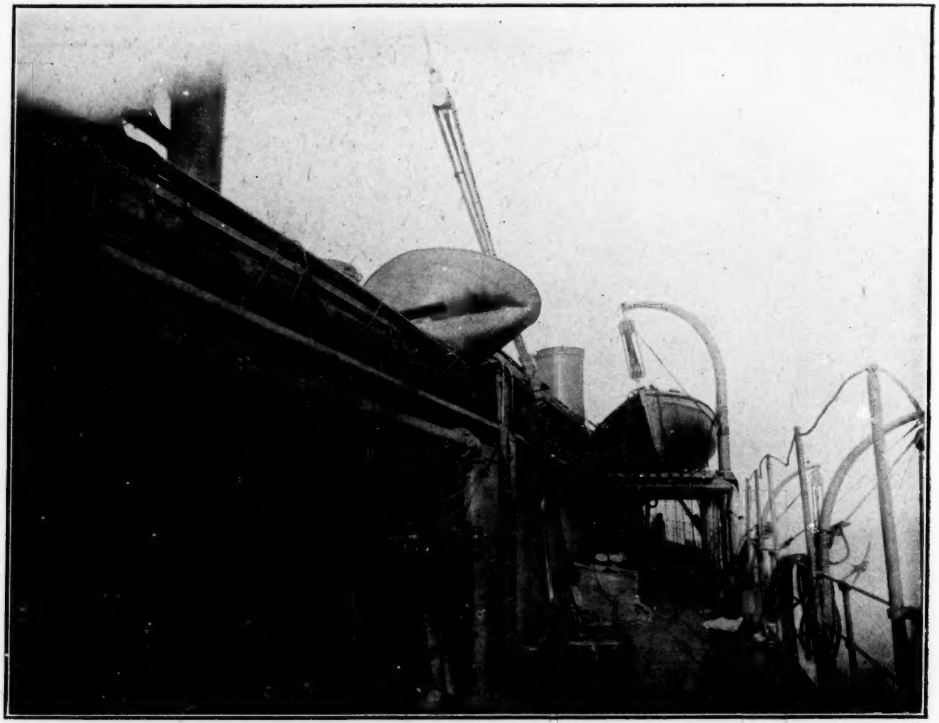
PORT QUARTER.



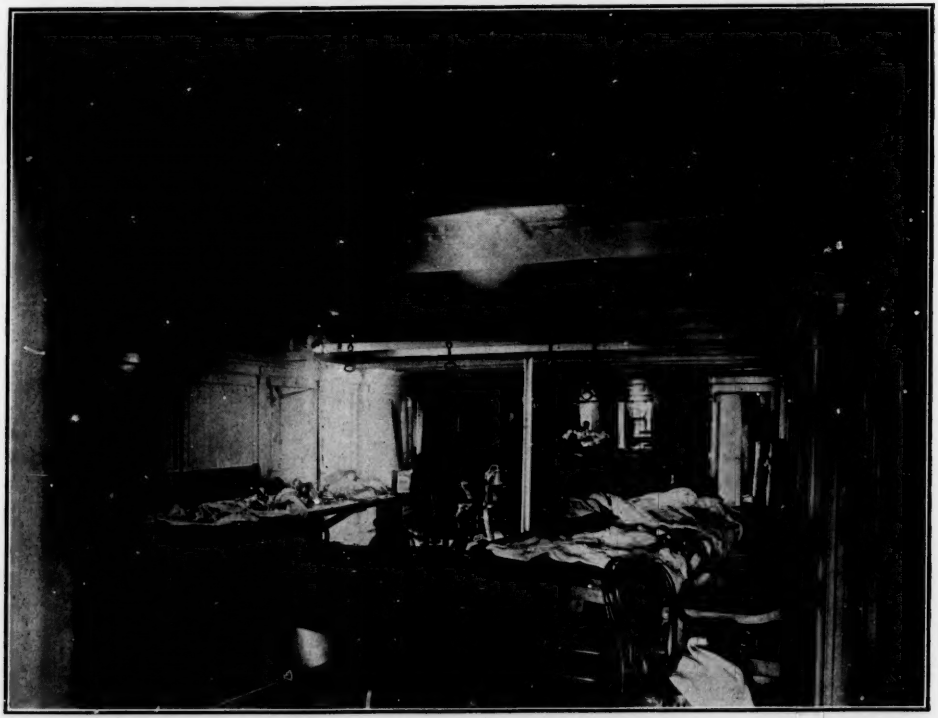
LOOKING FORWARD ON PORT SIDE AFT ABREAST OF SUPERSTRUCTURE.



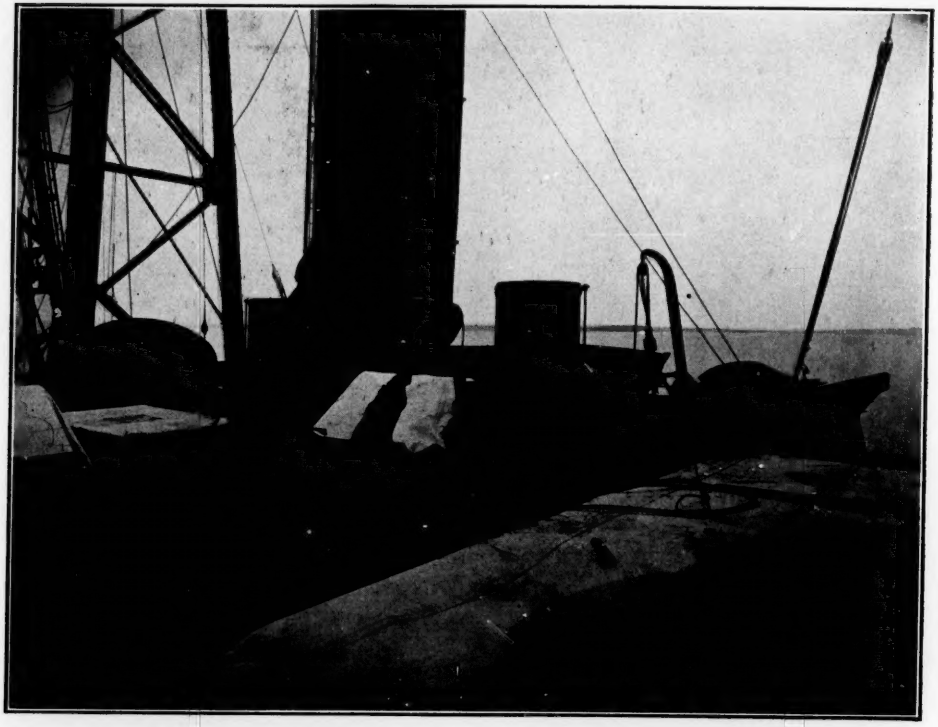
PORT SIDE OF SUPERSTRUCTURE—STEEL BULKHEAD.



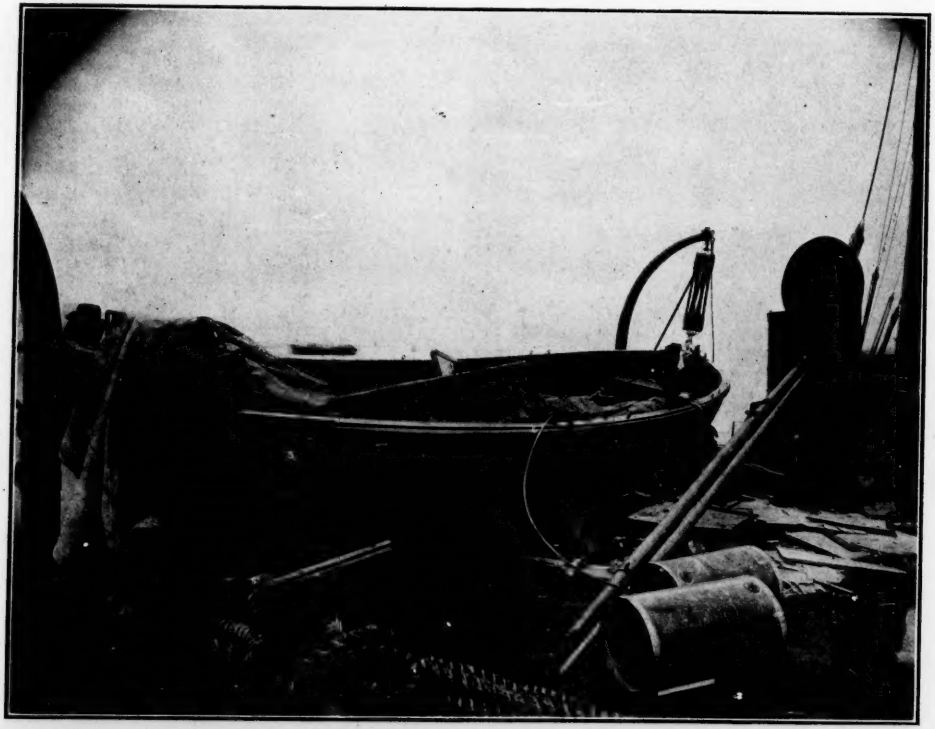
LOOKING AFT ALONG PORT SIDE OF SUPERSTRUCTURE AFT.



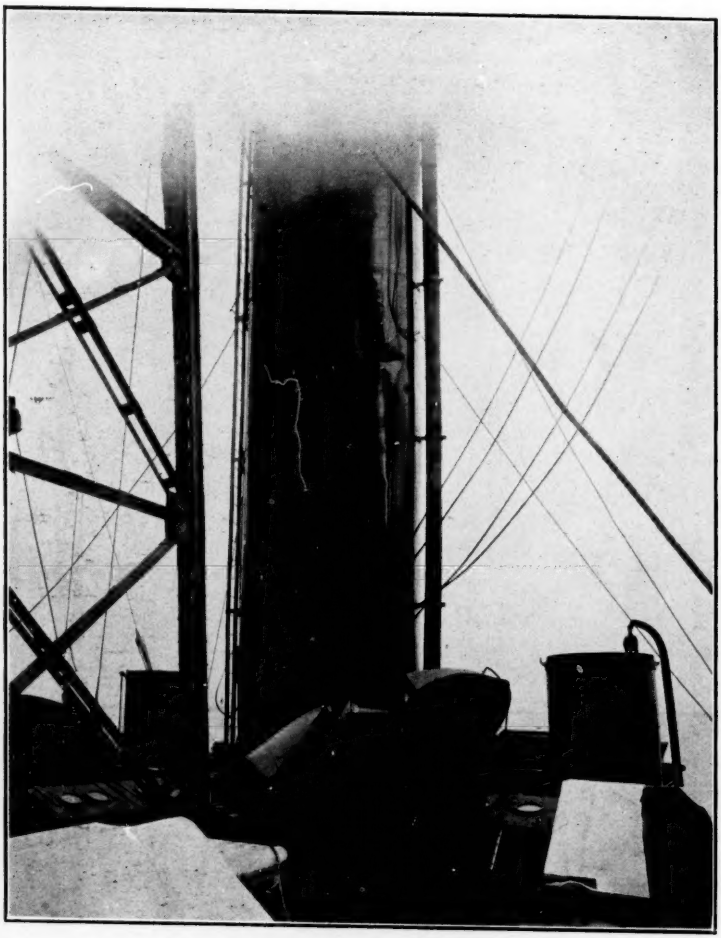
WARDROOM, LOOKING FORWARD.



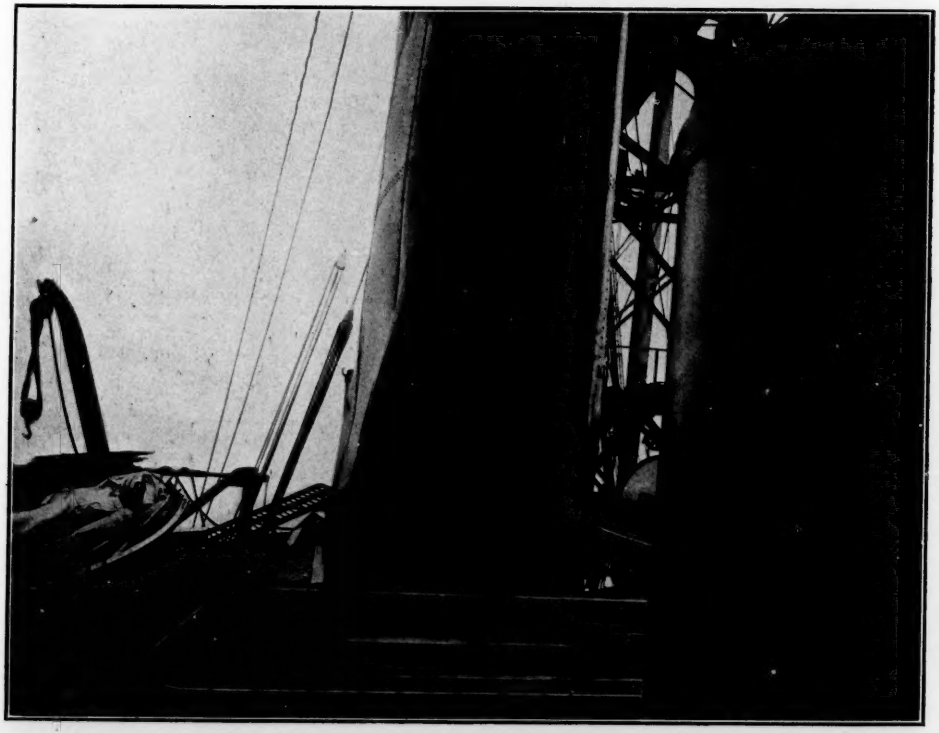
SUPERSTRUCTURE DECK. (AFT.)



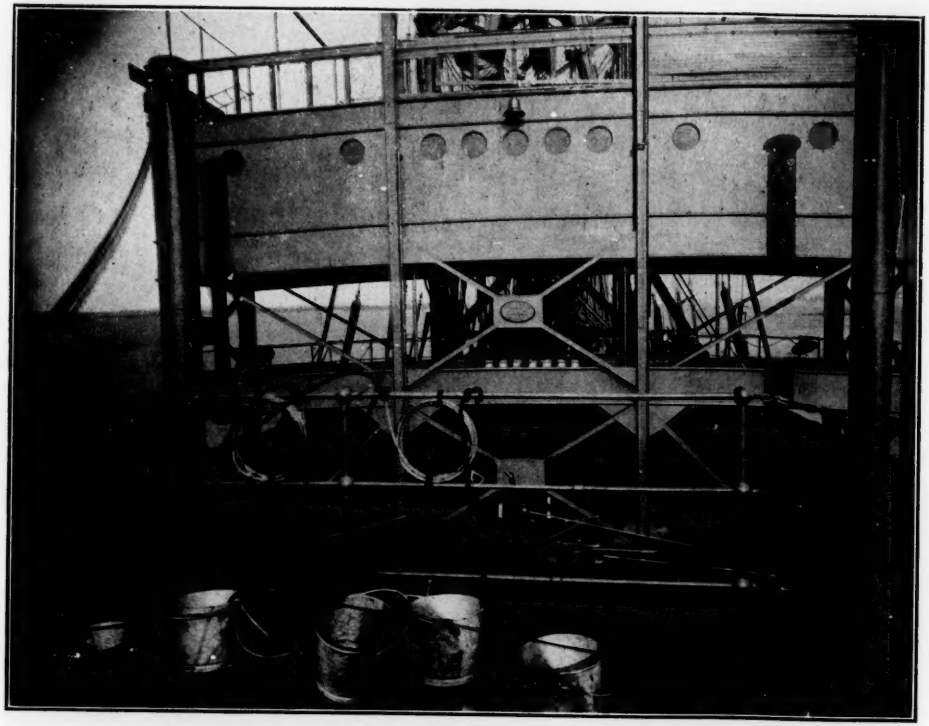
DAMAGED BOAT AND WRECKAGE ON SUPERSTRUCTURE.



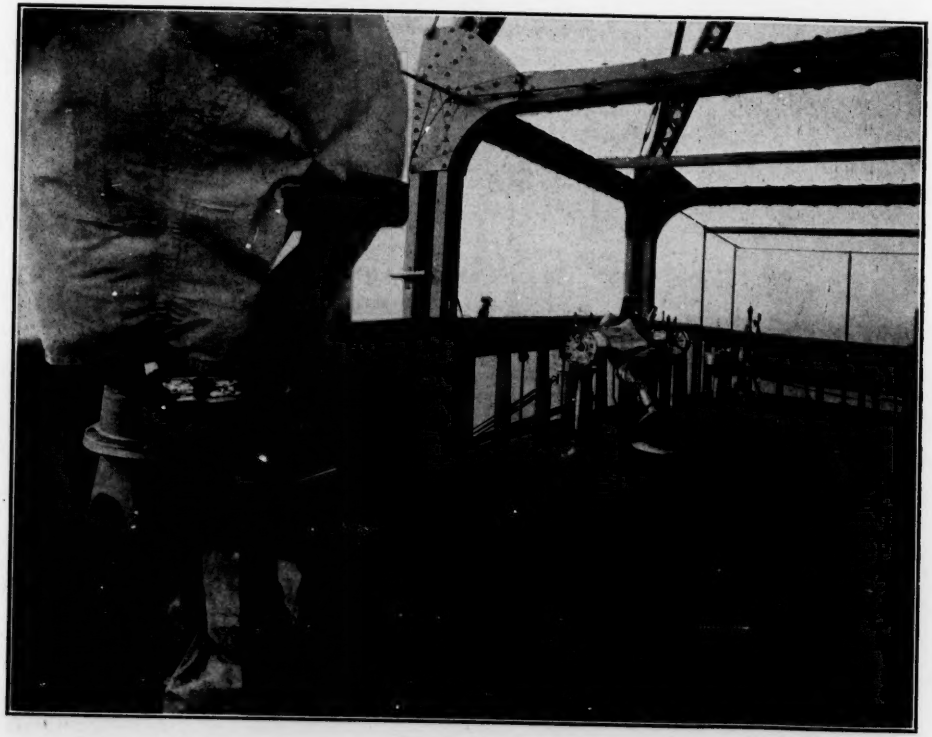
STARBOARD SMOKEPIPE FROM INBOARD.



PORT SMOKEPIPE AND LIFEBOAT FROM AFT.



LOOKING AFT FROM BREAK OF FORECASTLE.



BRIDGE.

BRIDGE CABIN, STARBOARD SIDE.

give a very incomplete idea of the actual damage that the *Jason* suffered. Had the vessel been without cargo instead of fully loaded, it is probable that the damage would have been much more severe, and it is probable that the entire hull would have been so severely strained as to make her acceptance by the Government doubtful. As it was, the hull was pierced in many places and so badly dented that it was necessary to replace a number of the side plates. The superstructure was badly wrenched and lopped over, the side bulkheads badly bulged, and the woodwork of the interior was a mass of splinters and wreckage. The deck of the superstructure was flattened and the supporting beams badly bent. The A frames of the coaling gear were pierced in many places by flying metal and the rigging of same was cut to shreds. The Ardois was torn from the mast. The bridge instruments were more or less damaged and the chart house and bridge cabin were wrecked. The watertight covers to the cargo-oil deep tanks was badly sprung. The smoke pipes and atmospheric exhaust pipes were pierced in many places and so badly wrenched as to require renewal.

The largest hole in the hull plating was about twelve inches in diameter. One piece of metal about three inches in diameter passed through the steel side of the superstructure, killed a mess attendant, then passed in a straight line through a steel athwartship bulkhead, through a steel fore-and-aft bulkhead, and through a wooden door into a lavatory. Another man was killed in a stateroom by being disemboweled by an airport lens and rim. A few other deaths were caused on the *Jason* by flying metal. The concussion was so great that persons standing on deck were thrown violently from their feet. Subsequent to the explosion the vessel presented the appearance of having been through a bombardment of shrapnel and shells as large as six or eight inches in size. Persons on board the *Jason* at the time stated that the greater part of the flying wreckage seemed to pass entirely over the *Jason*. Great quantities of pieces of metal and torn rivets, and about half a bushel of unburned pieces of dynamite were picked up about

the decks of the *Jason*, all in a smoky, grimy condition. The unburned dynamite was promptly thrown overboard in order to prevent further damage, but many pieces of torn metal were carried away by the crew and others as souvenirs of the terrible catastrophe. The cargo of coal was discharged to naval colliers, and the vessel was laid up for a period of approximately three months undergoing repairs.

The machinery apparently suffered no damage, as it was well below the water line and was therefore well protected.

The vessel was fully insured by the contractors, as required by the Government, and the underwriters were required to meet all costs of repairs.

An account of the trial performance of the *Jason* appeared in the August, 1913, number of the JOURNAL.

A fuller account and description of the sister vessel, the *Orion*, appeared in the November, 1912, number of the JOURNAL.

# TEST OF 30-INCH KEITH FAN.

## MADE AT U. S. NAVAL EXPERIMENTAL STATION, ANNAPOLIS, MD.

REPORT OF TEST MADE BY LEO LOEB, ASSOCIATE.

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### I.—OBJECT OF TEST.

2. The object of the test was to determine the speed and discharge required to maintain pressures of 4, 5 and 6 inches within the test chamber against several blast areas, the power to drive the fan and mechanical efficiency corresponding to the several conditions. In addition to the above schedule of tests as requested by the exhibitor it was thought advisable to conduct one series of tests at constant speed to give the head-discharge charactertistics, and for this series a speed of 1,630 r.p.m. was selected, since that is one rated speed of the fan.

### II.—DESCRIPTION OF APPARATUS.

3. The unit tested is shown in general arrangement on Inclosure F, and is of the centrifugal type as manufactured by James Keith and Blackman Co., Ltd., London, England*. The fan consists of a cast-iron hub, A, bolted to a 5/16-inch bottom plate of mild steel, 33⅝ inches in diameter. There are 24 blades, made up of No. 12 gage plate and riveted to the bottom plate and the stiffening rings. These stiffening rings, of which there are three intermediate rings of No. 16 gage plate, and an inlet ring, of No. 12 gage plate, are spaced $2\frac{11}{32}$ inches face to face of plate and are $2\frac{7}{16}$ inches wide. The fan inlet is 30 inches in diameter and the top or inlet plate is 37¼ inches over all. The overall depth of blades is 11 inches, and the blades have a $32\frac{7}{16}$-inch tip diameter at the bottom

* These fans are handled in this country by the Terry Steam Turbine Co.

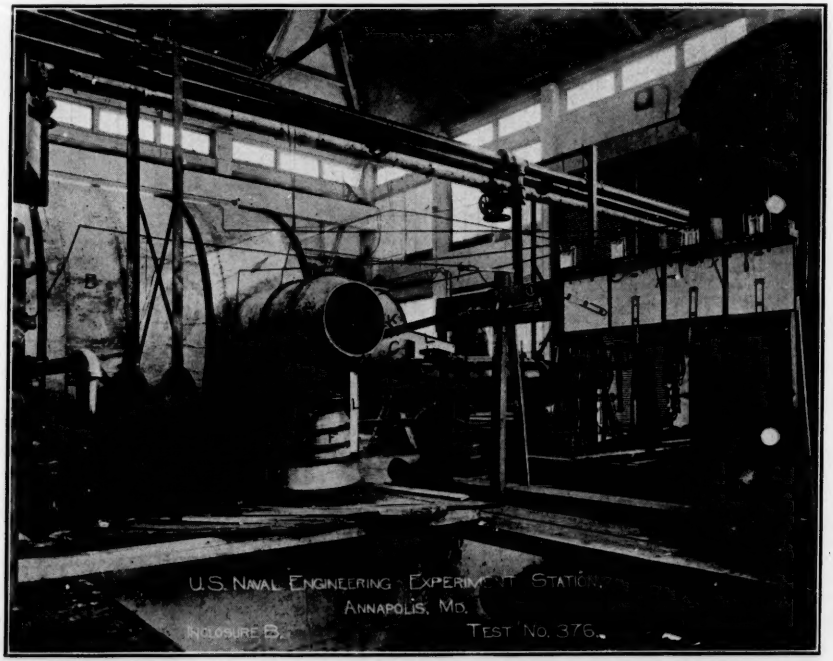
and a 36-inch tip diameter at inlet to fan. The blades taper from 10½ inches wide at the hub to 2½ inches where they terminate in the inlet ring. The fan as received was uncased, and the appearance of paint and finish indicated that the unit had seen service.

## III.—GENERAL ARRANGEMENT FOR TEST.

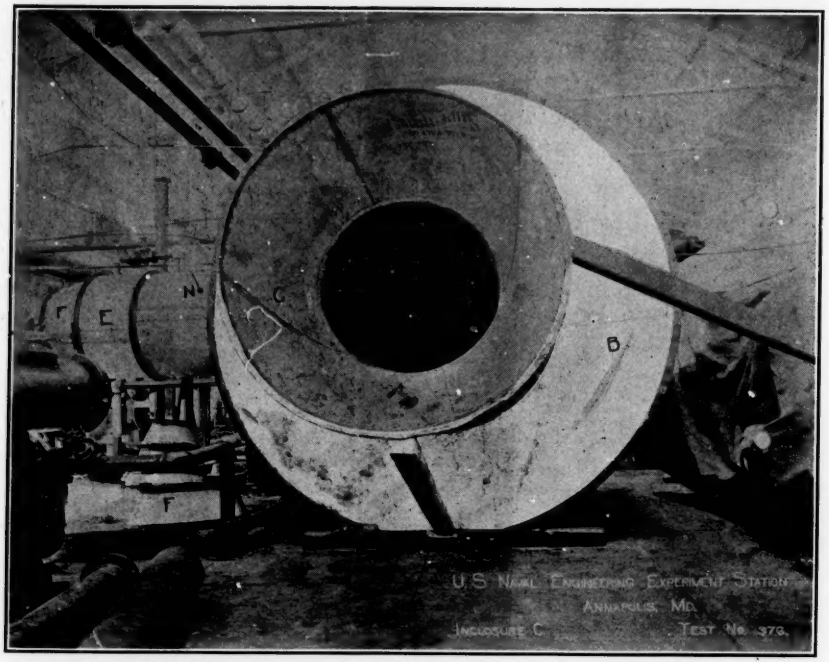
4. The general arrangement for test of the apparatus is shown in photographs, Inclosures (B), (C), (D) and (E). The fan was mounted horizontally on the end of the shaft of an electric dynamometer, A, and discharged into a test chamber, B. This chamber, ten feet in diameter and ten feet high, is constructed of 1/16-inch galvanized sheet metal. All joints are riveted and soldered and the structure is stiffened externally by four bands of 2½ × 2½-inch angle iron and internally by diagonal stays of flat iron. Air enters the inlet to the fan through the 60-inch funnel, C, and 37-inch suction duct, D, and is discharged through the 31½-inch duct, E, and any of the five nozzles, F. These nozzles are proportioned to give discharges of from 5,000 to 25,000 cubic feet per minute with a static house pressure of 5 inches. The stream lines at exit are made as nearly parallel as possible by an arrangement of baffles and screens in the discharge duct.

5. The electric dynamometer consists of a dynamo whose armature is carried in a floating frame. This frame is supported in ball-bearing pedestals, G, so that the torque to drive the machine as a motor is weighed on the scale beam, H. This torque includes the bearing and windage friction of the dynamometer as well as that to drive the fan. This machine friction was compensated by no-load runs of the dynamometer detached from the fan when operating at the same speeds as during the tests proper. The zero load obtained in this way varied only slightly between speeds of 1,000 and 1,800 r.p.m.

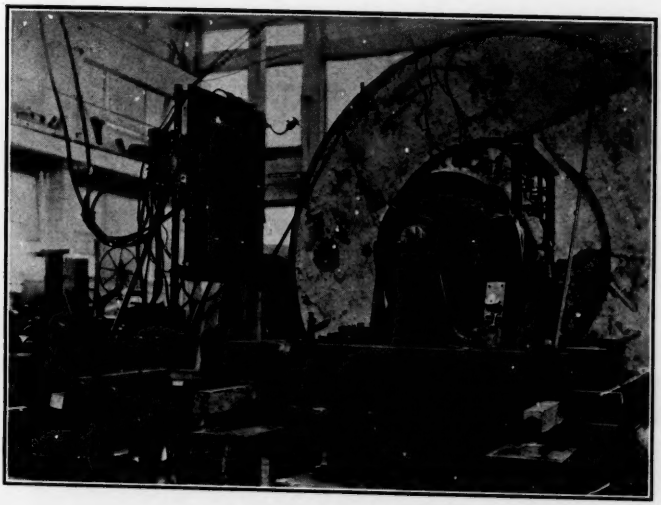
6. The speed of the blower was adjusted by reference to a hand tachometer applied to the end of the shaft, but more accurate count of total revolutions was obtained from a direct-reading Veeder counter attached at I and geared one hundred


U.S. NAVAL ENGINEERING EXPERIMENT STATION
ANNAPOLIS, MD.
INCLOSURE B.
TEST No. 376.

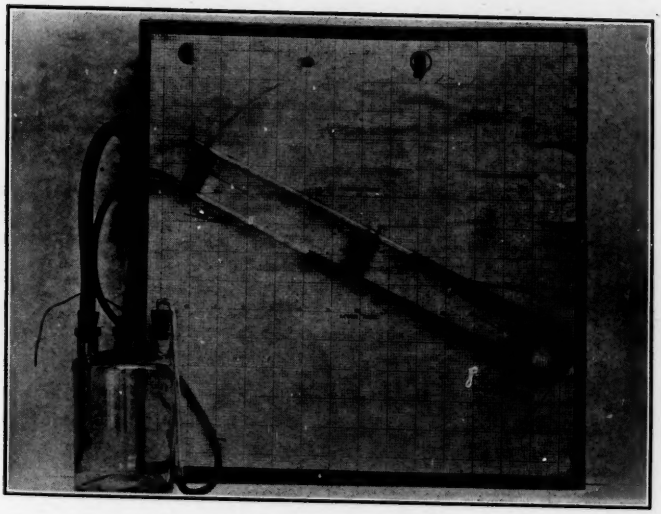
INCLOSURE (B).


E
C
B
F
U.S. NAVAL ENGINEERING EXPERIMENT STATION
ANNAPOLIS, MD.
INCLOSURE C
TEST NO. 372.

INCLOSURE (C).



INCLOSURE (D).



INCLOSURE (E).

to one and provided with an index for estimating to ten revolutions. By this means the limit of error in mean speed was three-tenths of a revolution per minute for a thirty-minute test.

7. Pressure at inlet to fan is shown by a simple impact tube mounted 12 inches ahead of the inlet and pointed upstream. This tube registers total pressure, both static and velocity, at inlet to the blower, and under favorable conditions should indicate a reading lower than atmospheric by only the friction in the suction duct. This pressure is registered on the differential gage, 6, Inclosure (B). The total static pressure of the fan discharge is taken at three points in the test house, distributed in such a way as to give pressure conditions at various heights and diameters within the compartment. At low rates of discharge the pressures shown by gages 3, 4 and 5 connected to these points were exactly in agreement, and at highest rate the maximum variation was only between 1/10 and 2/10 of an inch.

8. The measurements of volume of air discharged by the blower are made on the discharge side of the apparatus. As mentioned in paragraph 4, the air is released through nozzles of diameters 10.42 inches, 15.44 inches, 19.40 inches, 22.22 inches and 25.02 inches. The overall length of the discharge cone with all sections in place is 30 inches. Flush with the nozzle face are mounted two simple impact tubes, K, arranged to register on an inclined U-tube manometer, 2. The tubes are adjustable through stuffing-box guides to traverse the face of the nozzle horizontally and vertically and indicate by a pointer on graduated scales, L, the position of the center of the tube. The graduations on the scales correspond to the mean radii of ten annular rings of equal area for the several nozzle sections. Two horizontal and two vertical traverses are made during each run from which the average velocity head is computed.

9. A clearer view of the adjustable inclined U-tube manometer is shown in Inclosure (E). It consists of a U-tube constructed to permit the two parallel legs to always remain

in the same vertical plane, at the same time allowing them to take any desired inclination. The scale is a carefully selected sheet of cross-section paper. This manometer uses as a displacement fluid a mineral oil of density, 0.831, the advantage of oil over water being in the greater elevation for the same pressure and the elimination of capillary effects. The tube forms its own leveling device, since there is visible a distinct vertical and horizontal meniscus. The horizontal meniscus is leveled to the same horizontal line on the cross-section paper by swivelling the backing board, and readings are taken on the horizontal scale. The net reading multiplied by the inclination and the oil density is the true head. The gages are attached through Wolff bottles to eliminate vibration and fluctuations of oil level.

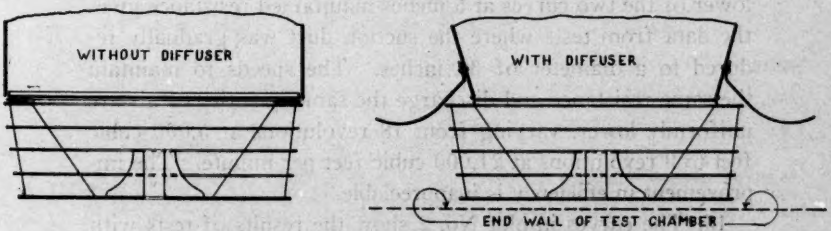
10. Temperature of air at inlet is taken at M by a thermometer projecting into the suction trunk and at discharge at N by a thermometer projecting into the discharge duct. Humidity calculations are made from observations on a wet and dry-bulb thermometer mounted near the blower. In order to get more nearly true wet-bulb depression a small fan is mounted to blow an ample supply of air against the wet-bulb thermometer.

## IV.—METHOD OF TESTING.

11. The suction trunk fitted to the test compartment was originally 31½ inches in diameter and it was proposed to connect this directly to the 30-inch fan inlet. However, the exhibitor requested that the suction duct be enlarged to 37 inches, which is the full fan diameter, and that the reduction take place at the inlet plate as shown in Inclosure F. Accordingly a new trunk was constructed and a series of tests at 4, 5 and 6 inches house pressure and at 1,631 r.p.m. was conducted at five rates of discharge. These tests as tabulated in Table I, runs 1 to 19, were in all cases but one 30 minutes in duration, with readings taken every five minutes. The speed of the fan was varied slightly from time to time during the test to maintain the exact pressure desired. Runs 1 to 5 are

at a nominal house pressure of 4 inches of water; runs 6 to 10 at 5 inches of water; runs 11 to 14 at 6 inches of water, and runs 15 to 19 at an average speed of 1,631 r.p.m

12. At the conclusion of these tests the suction trunk was altered as shown in arrangement with diffuser, Inclosure F, and the runs repeated. In this arrangement the suction duct tapers from the mean diameter of 37 inches to a diameter of 30⅜ inches at inlet, and the diffuser is built up of plate as shown. Runs 24 to 27 are with a nominal house pressure of 4 inches of water; 28 to 31 at 5 inches of water; 32 to 34 at 6 inches of water, and 35 to 38 at a mean speed of 1,631 r.p.m. There was such a decided improvement under these conditions from the open-fan installation that it was considered advis-




DETAILS OF SUCTION DUCT.—INCLOSURE (F).

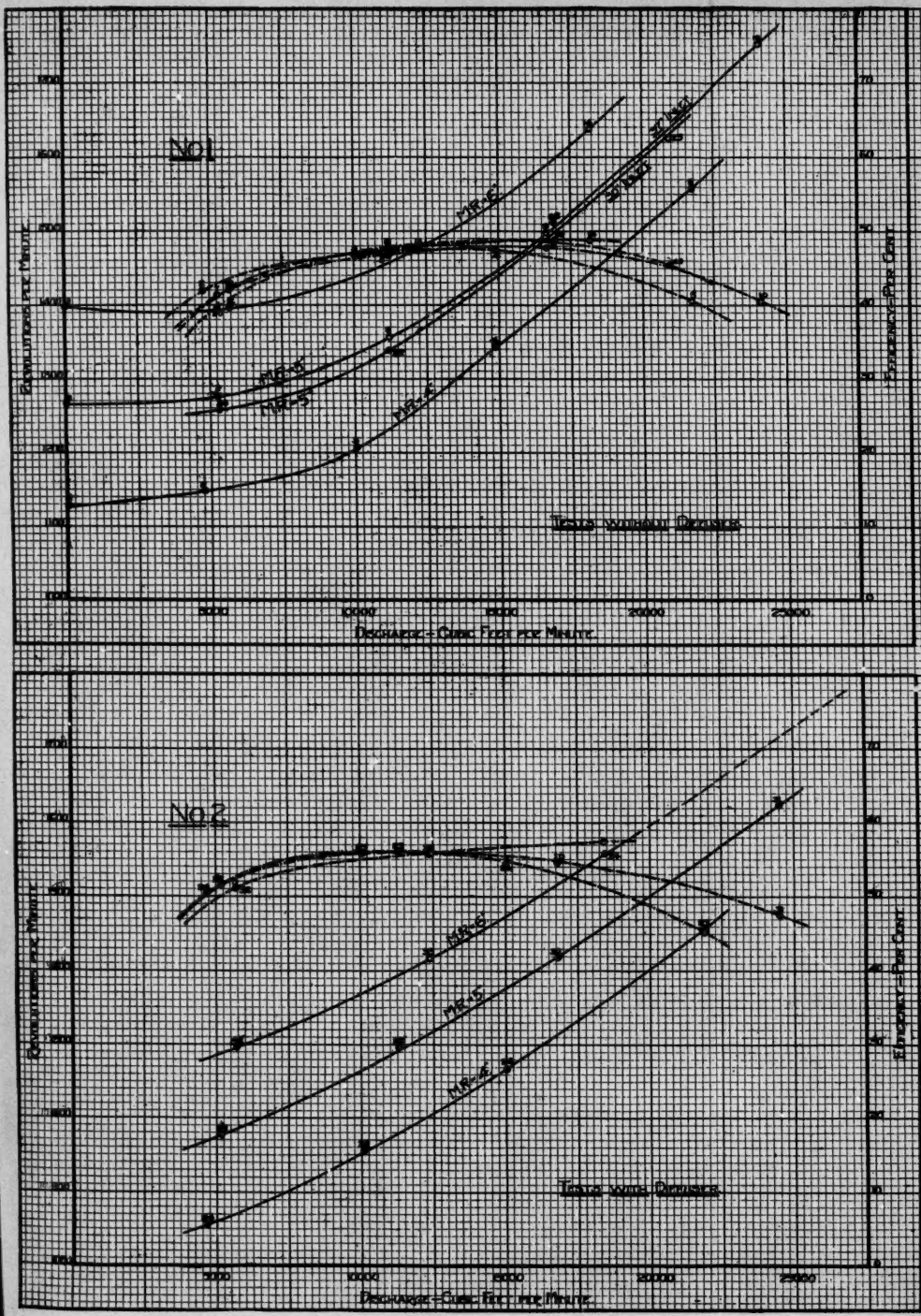
able to determine in how far the gain was due to diffuser and how far to gradually reduced inlet area. The diffuser was removed and four tests, runs 20 to 23, were made against a nominal house pressure of 5 inches of water as presented in Table I.

13. The procedure in each series of tests was: With any one discharge nozzle in position conduct a test at 4 inches pressure, then raise the house pressure by increasing the speed as required. The final run was at 1,631 r.p.m., after which the discharge area was changed. The data for runs 19 and 38 were calculated from the corresponding data of runs 14 and 31 by assuming that the discharge varied with the speed, the head with the square of the speed, and the mechanical efficiency remained constant for variations from 1,638 and 1,623 to desired of 1,631.

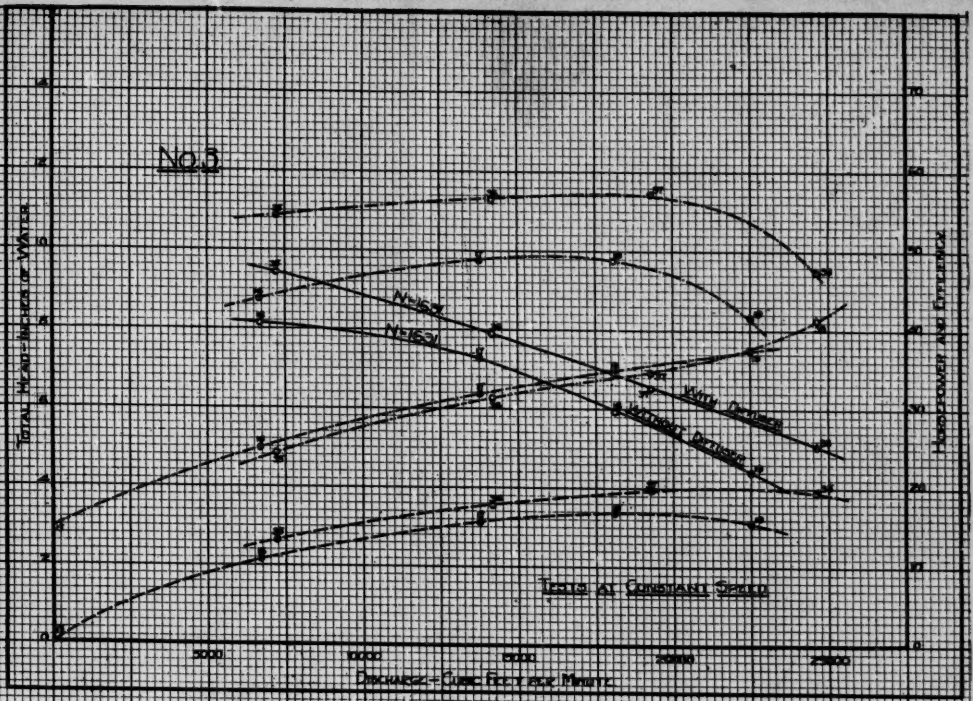
## V.—OBSERVATIONS AND DEDUCED RESULTS.

14. The data of Tables I and II are presented graphically in the fan performance curves of Drawing No. 474 and the design curves of Drawing No. 476. Plot No. 1 shows the results of tests without diffuser. The full lines represent the variation of revolutions per minute and quantity when the pressure is constant at 4, 5 and 6 inches. The broken lines are the corresponding mechanical efficiencies. For reduced capacity the quantity increases in greater proportion than the speed, but above 10,000 cubic feet per minute at 4 inches, 11,250 cubic feet per minute at 5 inches and 12,250 cubic feet per minute at 6 inches the quantity increases in almost direct proportion to the speed within the limits of the test. The lower of the two curves at 5 inches maintained resistance gives the data from tests where the suction duct was gradually reduced to a diameter of 30 inches. The speeds to maintain the same resistance and discharge the same quantity of air are uniformly lower, varying from 18 revolutions at 5,000 cubic feet to 9 revolutions at 21,000 cubic feet per minute. The improvement in efficiency is inappreciable.

15. The curves in plot No. 2 show the results of tests with diffuser. Although the tests were not carried down to the condition of closed dampers with no fan discharge, there is a very evident improvement in the general trend of the curves. At 5 inches discharge the revolutions to give 5,000 and 25,000 cubic feet per minute discharge are 1,173 and 1,638 as against 1,272 and 1,785 for open fan, and the mechanical efficiencies corresponding are 51.0 per cent. and 46.9 per cent. as against 40.2 per cent. and 39.8 per cent. for open fan. The reduction in speed as insuring more reliable operation and reduced repairs is of even greater importance than the gain in efficiency. The tests at six inches were not carried to the largest nozzle, but using the same ratios as obtained with greatest discharges and 5 inches house pressure the speed to maintain 6 inches would be 1,778 r.p.m., the discharge 26,820 cubic feet per minute and the shaft horsepower 53.3.


No 1
No 2
Revolutions per Minute
Efficiency - Per Cent
Total Head - Inches of Water
Tests without Orifice
Tests with Orifice
Discharge - Cubic Feet per Minute
15000
20000
25000
70
60
50
40
30
20
10

Inclosure (



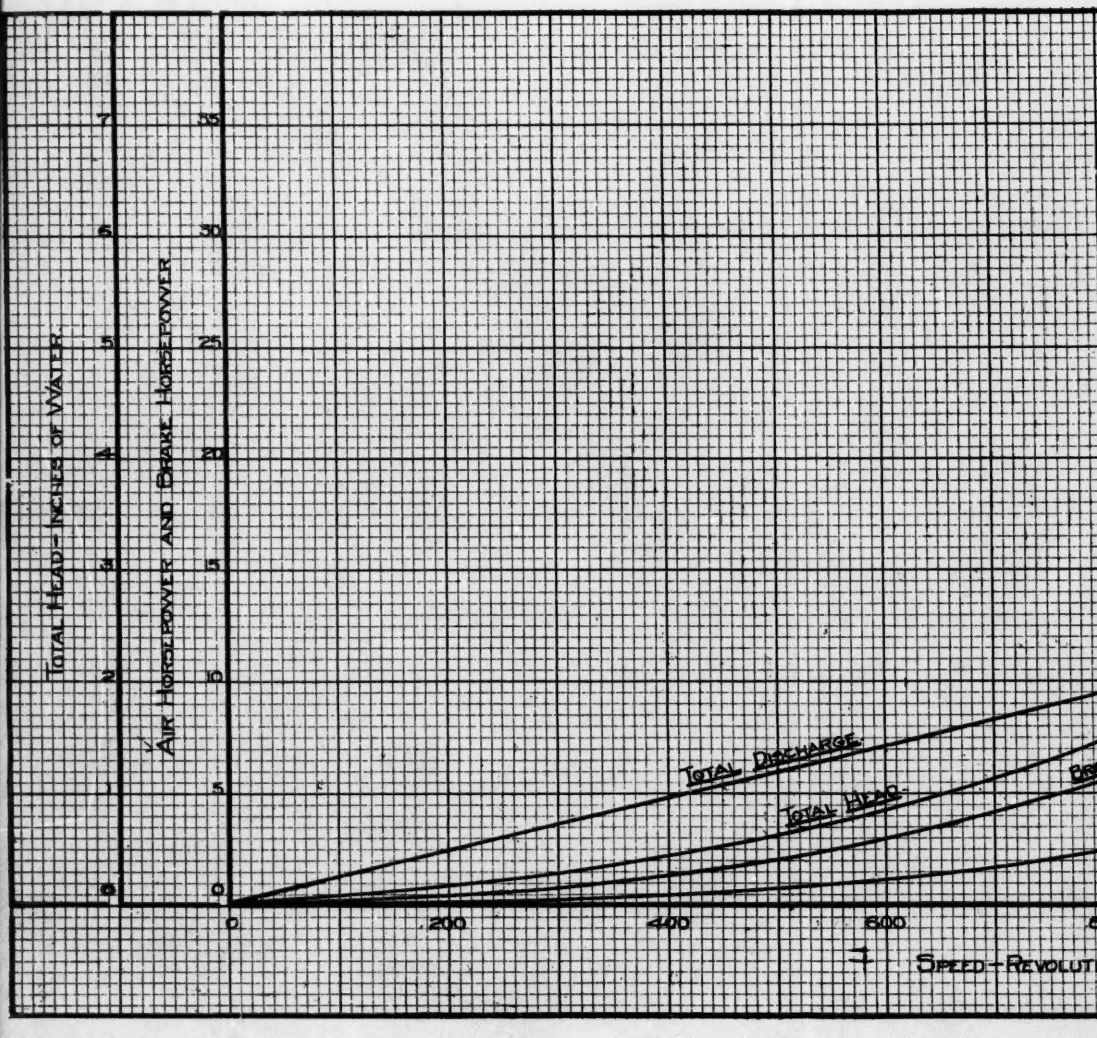

Note:—
Full Lines Indicate No.1 and No.2 Revolutions per Minute; No.3 Maintained Pressure.
Broken Lines Indicate No.3 Air Horsepower.
Broken Lines with Dot Indicate Mechanical Efficiency.
Broken Lines with Two Dots Indicate No.3 Brake Horsepower.
N=Speed Revolutions per Minute.
MR=Maintained Resistance in Test Chamber.
Small Numerals Indicate Number of Run.

FAN PERFORMANCE CURVES
30-INCH KEITH FAN
TEST 376

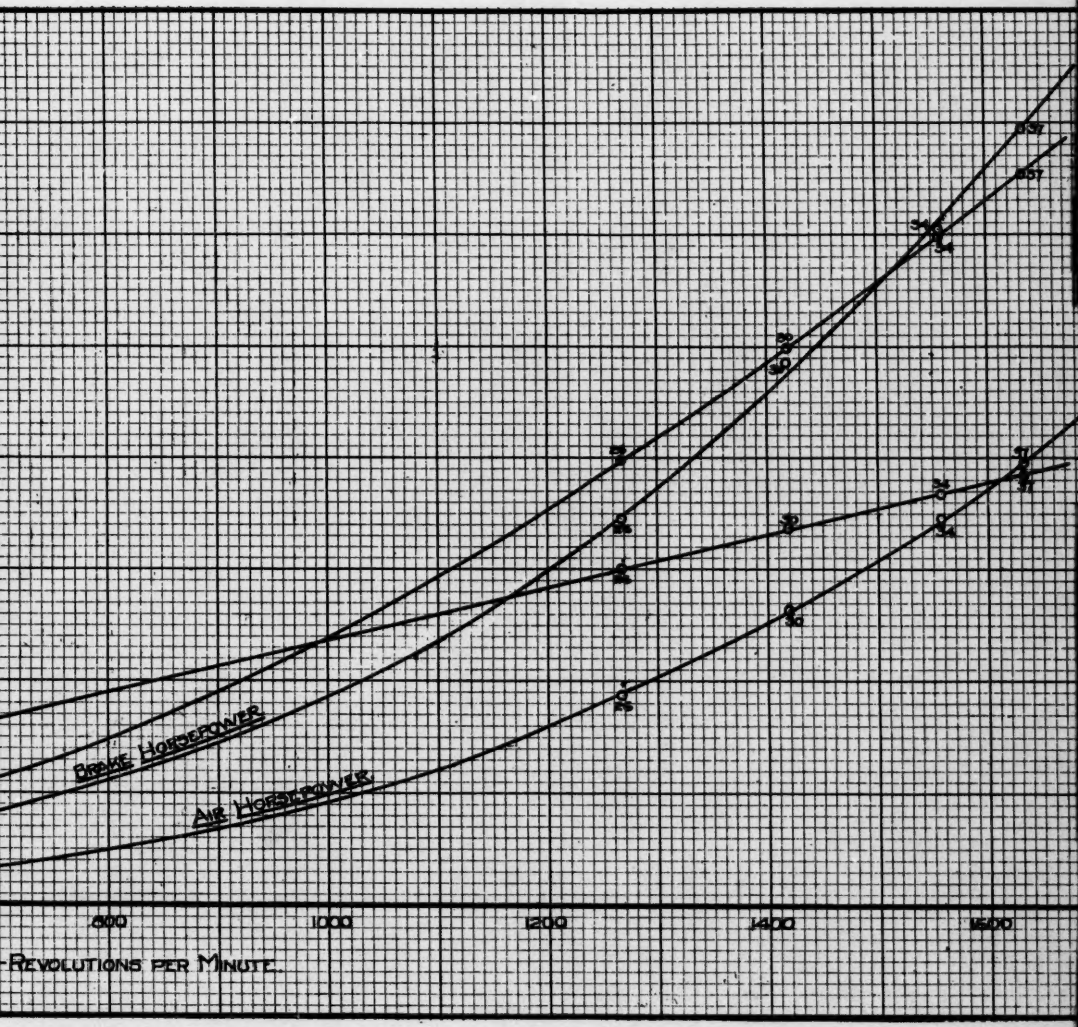
Drawn by A.W. Traced by A.W. Checked by H.L. Date Nov 6 1920
U.S. NAVAL ENGINEERING EXPERIMENT STATION
ANNAPOLIS MD

DF No. 474
Case 2, Drawer 22

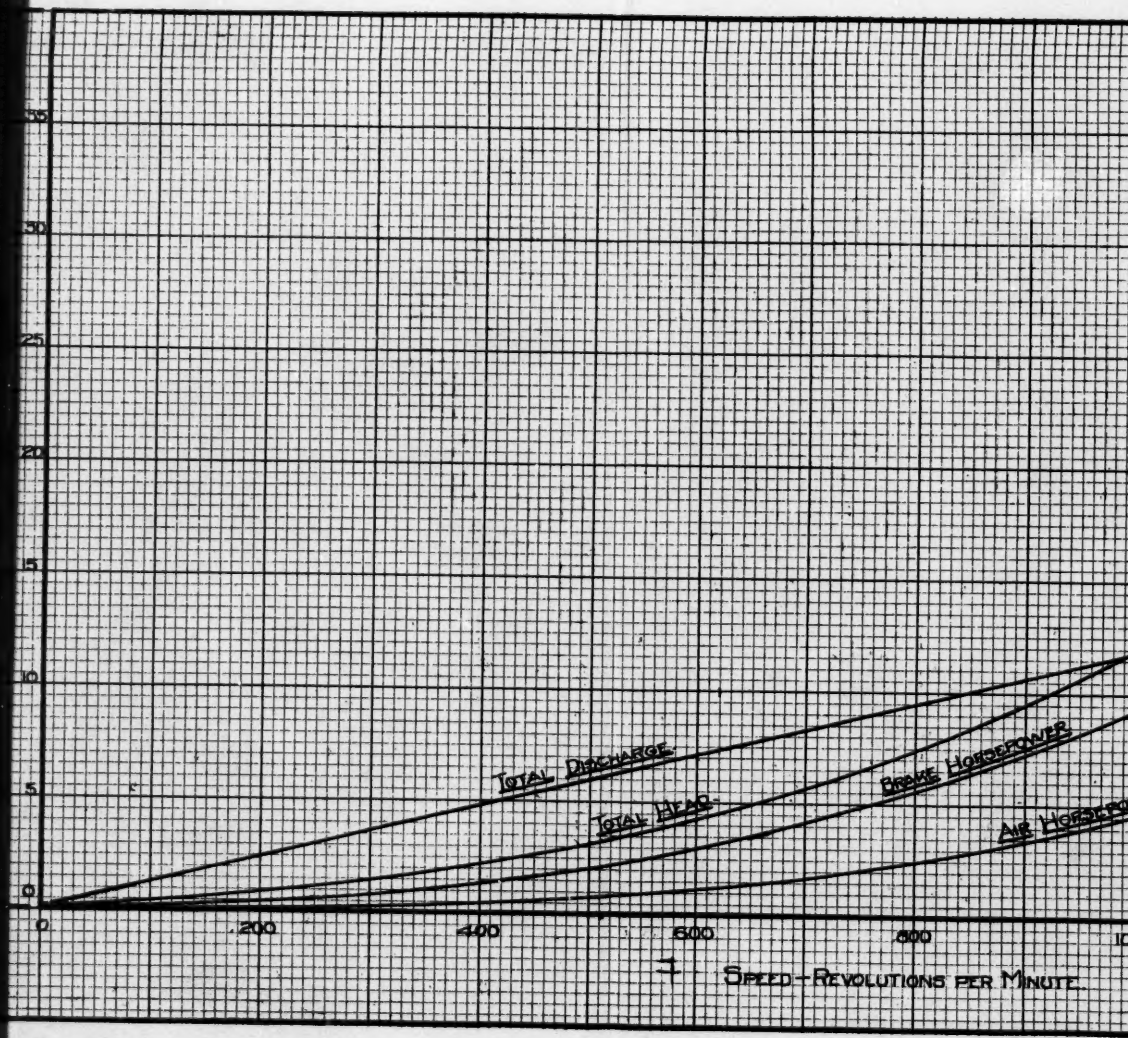
LOSURE (I).



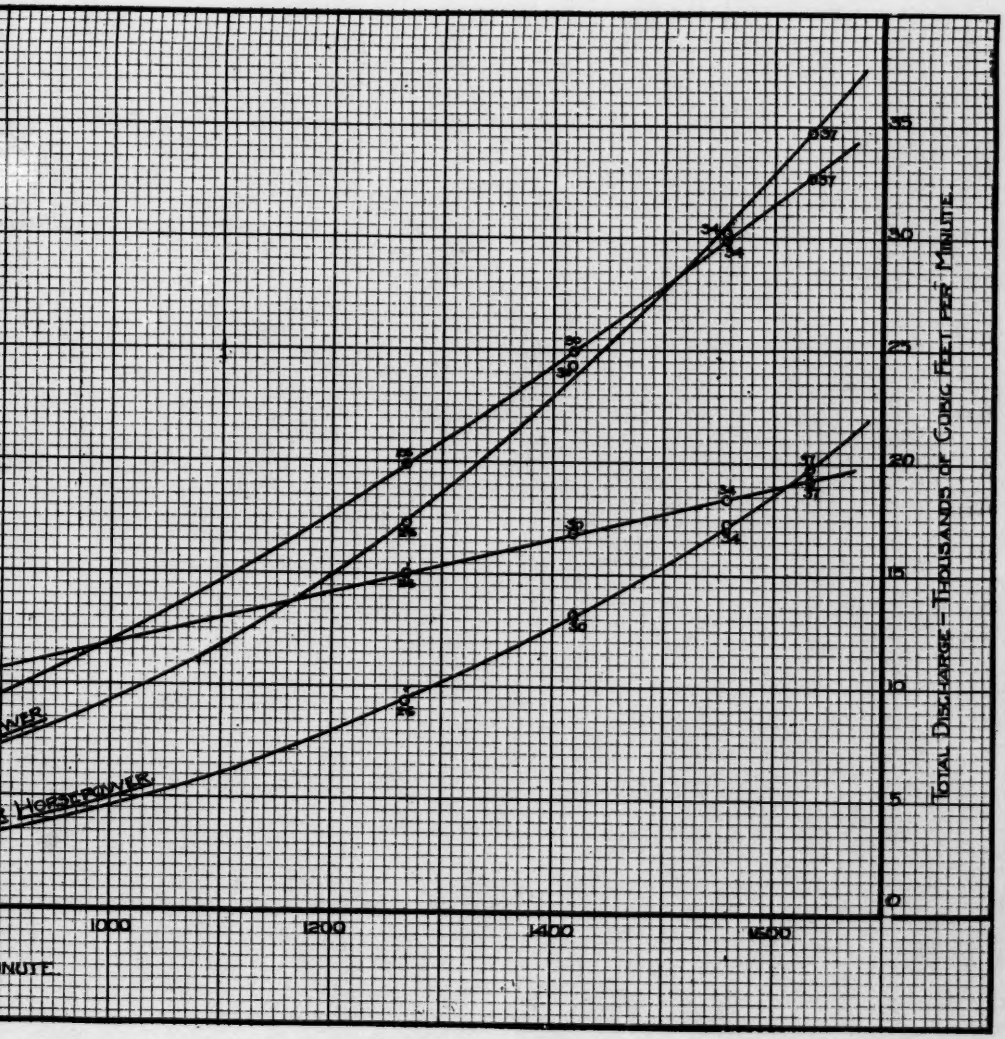

Fan Characteristics, 30-Inc




CS, 30-INCH KEITH FAN.—INCLOSURE (J).




FAN CHARACTERISTICS, 30-INCH KEITH FAN.—INC




FAN.—INCLOSURE (J).

TABLE I.—TEST OF 30-INCH KEITH FAN.

TABLE OF DATA AND RESULTS. TESTS WITHOUT DIFFUSER.

| | General data | | | | Air data. | | | | | | | | | | Power. | | Deduced data. | | | | | 37-inch inlet. | 30-in.in. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Number of test. | Date, October-November, 1913 | Duration of test, minutes. | Barometer, inches of mercury. | Revolutions per minute. | Room temperature, degrees Fahrenheit. | Air temperature in suction duct, degrees F. | Air temperature at discharge, degrees F. | Relative humidity, per cent. | Total pressure at inlet to fan, inches of water suction. | Static pressure at outlet of fan, inches of water. | Total discharge pressure, inches of water. | Average velocity head at nozzle, inches of water. | Density of air, pounds per cubic foot. | Nozzle velocity, feet per minute. | Nozzle area, square feet. | Nozzle discharge, cubic feet per minute. | Brake load, pounds. | Air horsepower. | Brake horsepower. | Mechanical efficiency, per ct. | Tip-speed pressure, inches of water. | Fan displacement, cubic feet per minute. | Ratio: $\frac{M.R.}{T.S.P.}$ | Ratio: $\frac{\text{Actual Discharge}}{\text{Fan Displacement}}$ | | |
| 1 | 29 | 30 | 29.91 | 1,127 | 71.8 | 77.1 | 78.3 | 73 | 2.000 | 3.959 | 5.959 | .07311 | 8,063 | 9,675 | .592 | 5,254 | 16.38 | .00 | 6.13 | .... | 6.199 | 6,597 | .639 | .000 | .000 | 0.713 |
| 2 | 29 | 30 | 29.78 | 1,140 | 73.6 | 71.4 | 77.0 | 73 | 2.110 | 3.982 | 5.992 | .07305 | 44,773 | 9,675 | .592 | 44,196 | 23.77 | 3.83 | 9.10 | 42.0 | 6.443 | 6,726 | .619 | .713 | .964 | 1.448 |
| 3 | 30 | 30 | 29.98 | 1,207 | 70.3 | 70.1 | 73.8 | 45 | .010 | 3.985 | 3.995 | .07423 | 10,008 | 7,708 | 1.301 | 10,008 | 33.66 | 6.30 | 13.54 | 46.6 | 7.319 | 7,066 | .552 | 1.425 | 1.809 | 1.425 |
| 4 | 30 | 30 | 29.97 | 1,344 | 71.8 | 70.5 | 74.2 | 45 | .004 | 3.981 | 3.987 | .07409 | 14,894 | 7,301 | 2.040 | 14,894 | 43.97 | 9.33 | 19.70 | 47.5 | 8.936 | 7,868 | .445 | 1.869 | 2.385 | 1.869 |
| 5 | 30 | 30 | 30.25 | 1,558 | 68.7 | 67.8 | 71.8 | 32 | .003 | 3.984 | 3.987 | .07525 | 21,678 | 6,410 | 3.382 | 21,678 | 64.58 | 13.60 | 33.54 | 40.5 | 12.188 | 9,121 | .327 | 2.385 | .000 | 2.385 |
| 6 | 30 | 30 | 29.85 | 1,267 | 73.9 | 80.7 | 83.3 | 73 | 3.092 | 4.980 | 8.072 | .07221 | .... | .000 | .592 | .... | 18.07 | .00 | 7.63 | .... | 7.712 | 7,417 | .644 | .000 | .644 | .704 |
| 7 | 29 | 30 | 29.77 | 1,475 | 71.5 | 71.5 | 77.3 | 73 | 1.380 | 4.980 | 6.360 | 4.772 | .07300 | 8,861 | .592 | 5,246 | 30.01 | 5.85 | 12.75 | 41.2 | 7.924 | 7,464 | .629 | .704 | .551 | 1.411 |
| 8 | 30 | 30 | 29.97 | 1,357 | 70.8 | 70.6 | 75.0 | 44 | .007 | 5.015 | 5.022 | 4.574 | .07406 | 8,609 | 1.301 | 11,200 | 41.82 | 8.85 | 18.92 | 46.8 | 9.101 | 7,944 | .551 | 1.411 | 1.698 | 1.698 |
| 9 | 30 | 30 | 29.97 | 1,498 | 70.9 | 70.1 | 74.6 | 43 | .005 | 4.990 | 4.995 | 4.101 | .07415 | 8,154 | 2.040 | 16,635 | 54.75 | 13.07 | 27.34 | 47.8 | 11.110 | 10,256 | .449 | 1.698 | 2.358 | 2.358 |
| 10 | 30 | 20 | 30.28 | 1,752 | 67.7 | 66.9 | 71.8 | 33 | .004 | 5.001 | 5.005 | 3.304 | .07531 | 7,153 | 3.382 | 24,191 | 80.63 | 19.05 | 47.09 | 40.5 | 15.428 | 8,184 | .324 | 2.358 | .000 | .... |
| 11 | 29 | 30 | 29.83 | 1,398 | 73.0 | 81.6 | 86.9 | 73 | .3824 | 5.983 | 9.807 | .... | .07127 | .... | .000 | .... | 21.90 | .00 | 10.08 | .... | 9.998 | 8,184 | .643 | .000 | .643 | .... |
| 12 | 30 | 30 | 29.97 | 1,394 | 68.7 | 68.9 | 75.4 | 47 | 1.630 | 5.977 | 7.607 | 5.761 | .07401 | 9,672 | .592 | 5,726 | 35.03 | 6.85 | 16.28 | 42.1 | 9.595 | 8,160 | .623 | .702 | .702 | .702 |
| 13 | 30 | 30 | 29.98 | 1,478 | 71.4 | 71.2 | 76.3 | 48 | .005 | 5.981 | 5.947 | 5.453 | .07387 | 9,418 | 1.301 | 12,253 | 48.86 | 11.50 | 24.07 | 47.8 | 10.776 | 8,852 | .553 | 1.418 | 1.418 | 1.418 |
| 14 | 30 | 30 | 29.98 | 1,636 | 70.4 | 69.6 | 74.9 | 45 | .005 | 5.974 | 5.979 | 4.878 | .07407 | 8,888 | 2.040 | 18,132 | 64.02 | 17.06 | 34.96 | 48.8 | 13.974 | 9,389 | .451 | 1.892 | .000 | 1.892 |
| 15 | 29 | 30 | 29.83 | 1,631 | 73.3 | 83.9 | 95.9 | 73 | 5.125 | 5.052 | 13.177 | .... | .07044 | .... | .000 | .... | 27.11 | .00 | 14.74 | .... | 12.599 | 9,548 | .644 | .000 | .644 | .... |
| 16 | 30 | 30 | 29.98 | 1,629 | 71.0 | 71.0 | 80.2 | 46 | 2.100 | 8.151 | 10.951 | 7.810 | .07335 | 11,310 | .592 | 6,696 | 45.43 | 10.80 | 24.67 | 43.8 | 12.994 | 9,536 | .687 | .703 | .703 | .703 |
| 17 | 30 | 30 | 29.98 | 1,632 | 72.0 | 71.6 | 78.0 | 40 | .008 | 7.274 | 7.281 | 6.599 | .07361 | 10,380 | 1.301 | 13,504 | 58.24 | 13.47 | 31.68 | 42.8 | 13.092 | 9,554 | .556 | 1.413 | 1.413 | 1.413 |
| 18 | 30 | 30 | 29.97 | 1,631 | 69.6 | 69.6 | 74.9 | 45 | .005 | 5.923 | 4.831 | .07407 | 8,855 | 2.040 | 18,065 | 63.55 | 16.87 | 34.59 | 48.8 | 13.154 | 9.548 | .451 | 1.892 | .000 | 1.892 |
| 19 | 30 | 30 | 30.09 | 1,631 | 67.7 | 67.0 | 71.1 | 32 | .001 | 4.397 | 4.398 | 2.786 | .07549 | 6,653 | 3.382 | 22,500 | 69.04 | 15.57 | 37.53 | 41.5 | 13.394 | 9,548 | .328 | 2.358 | .000 | 2.358 |
| 20 | 7 | 30 | 30.12 | 1,258 | 70.9 | 69.8 | 75.1 | 57 | 1.013 | 4.998 | 6.011 | 4.860 | .07424 | 8,875 | .592 | 5,254 | 29.95 | 4.97 | 12.56 | 39.6 | 7.841 | 7,364 | .637 | 0.713 | 0.713 | 0.713 |
| 21 | 7 | 30 | 30.13 | 1,339 | 73.5 | 73.4 | 77.3 | 56 | .003 | 4.987 | 4.990 | 4.583 | .07301 | 8,608 | 1.301 | 11,196 | 41.11 | 8.79 | 18.35 | 47.9 | 8.847 | 7,839 | .564 | 1.448 | .964 | 1.448 |
| 22 | 7 | 30 | 30.08 | 1,518 | 76.1 | 75.7 | 79.7 | 53 | .000 | 4.990 | 4.990 | 4.200 | .07354 | 8,284 | 2.040 | 16,899 | 53.91 | 13.27 | 27.17 | 48.8 | 11.225 | 8,951 | .445 | 1.909 | 1.909 | 1.909 |
| 23 | 7 | 30 | 30.06 | 1,625 | 78.6 | 77.7 | 81.0 | 58 | .000 | 4.967 | 4.967 | 3.673 | .07330 | 7,756 | 2.694 | 20,895 | 66.49 | 16.33 | 36.02 | 45.3 | 12.911 | 9,513 | .385 | 2.196 | .000 | 2.196 |

TABLE II.—TEST OF 30-INCH KEITH FAN.

TABLE OF DATA AND RESULTS. TEST WITH DIFFUSER. 30-INCH INLET.

| | General data. | | | Air data. | | | | | | | | | | | Power. | | Deduced data. | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 |
| Number of test. | Date, November, 1913. | Duration of test, minutes. | Barometer, inches of mercury. | Revolutions per minute. | Room temperature, degrees Fahrenheit. | Air temperature, in suction duct, degrees Fahrenheit. | Air temperature at discharge, degrees Fahrenheit. | Relative humidity, per cent. | Total pressure at inlet to fan, inches of water suction. | Static pressure at outlet of fan, inches of water. | Total discharge pressure, inches of water. | Average velocity head at nozzle, inches of water. | Density of air, pounds per cubic foot. | Nozzle velocity, feet per minute. | Nozzle area, square feet. | Nozzle discharge, cubic feet per minute. | Brake load, pounds. | Air horsepower. | Brake horsepower. | Mechanical efficiency, per cent. | Tip-speed pressure, inches of water. | Fan displacement, cubic feet per minute. | Ratio: $\frac{M.R.}{T.S.P.}$ | Ratio: $\frac{\text{Actual Discharge}}{\text{Fan Displacement}}$. |
| 24 | 6 | 30 | 30.27 | 1,056 | 67.8 | 67.8 | 71.0 | 56 | .871 | 3.958 | 4.899 | 3.901 | .07532 | 7,890 | 0.592 | 4,671 | 20.01 | 3.55 | 7.04 | 50.4 | 5.669 | 6,182 | .706 | .756 |
| 25 | 5 | 30 | 30.24 | 1,155 | 71.4 | 71.4 | 74.3 | 48 | .019 | 3.974 | 3.993 | 3.710 | .07473 | 7,783 | 1.301 | 10,048 | 29 32 | 6.31 | 11.29 | 55.9 | 6.657 | 6,761 | .597 | 1.486 |
| 26 | 5 | 30 | 30.29 | 1,268 | 67.8 | 67.6 | 70.3 | 44 | .002 | 3.983 | 3.985 | 3.399 | .07550 | 7,355 | 2.040 | 15,004 | 40.69 | 9.41 | 17.80 | 54.7 | 8.100 | 7,423 | .492 | 2.021 |
| 27 | 6 | 30 | 30.00 | 1,458 | 70.3 | 70.0 | 73.4 | 47 | .000 | 3.991 | 3.991 | 2.581 | .07438 | 6,456 | 3.382 | 21,834 | 62.41 | 13.71 | 30.33 | 45 2 | 10.544 | 8,535 | .379 | 2.358 |
| 28 | 6 | 30 | 30.28 | 1,177 | 70.7 | 70.8 | 74.7 | 52 | 1.068 | 4.990 | 6.058 | 4.831 | .07476 | 8,814 | 0.592 | 5,218 | 24.60 | 4.97 | 9.65 | 51.5 | 6 911 | 6,890 | .722 | .757 |
| 29 | 5 | 30 | 30.24 | 1,297 | 72.4 | 71.7 | 75.7 | 42 | .025 | 5.034 | 5.059 | 4.690 | .07458 | 8,694 | 1.301 | 11,311 | 37.08 | 9.00 | 16.03 | 56.1 | 8.376 | 7,593 | .601 | 1.490 |
| 30 | 5 | 30 | 30.29 | 1,419 | 70.4 | 70.0 | 73.2 | 40 | .001 | 4.995 | 4.996 | 4.268 | .07512 | 8,254 | 2.040 | 16,838 | 51.31 | 13.24 | 24.47 | 54 6 | 10.102 | 8,307 | .495 | 2.027 |
| 31 | 6 | 24 | 30.00 | 1,683 | 72.0 | 71.3 | 75.4 | 45 | .000 | 4.990 | 4.990 | 3.233 | .07403 | 7,244 | 3.382 | 24,499 | 74.92 | 19.24 | 40.53 | 47.5 | 13.015 | 9.501 | .383 | 2.579 |
| 32 | 5 | 30 | 30.27 | 1,296 | 71.2 | 70.7 | 75.7 | 47 | 1.316 | 5.097 | 7.313 | 5.810 | .07468 | 9,667 | 0.592 | 5,723 | 29.82 | 6.58 | 12.88 | 51.1 | 8.371 | 7,587 | .716 | .754 |
| 33 | 5 | 30 | 30.24 | 1,417 | 72.9 | 72.5 | 76.9 | 46 | .026 | 5.066 | 5.092 | 4.588 | .07436 | 9,452 | 1.301 | 12,297 | 43.99 | 11.59 | 20.78 | 55.8 | 9.068 | 8,295 | .599 | 1.482 |
| 34 | 5 | 30 | 30.28 | 1,558 | 72.9 | 72.5 | 76.1 | 38 | .000 | 5.075 | 5.075 | 5.080 | .07464 | 9,045 | 2.040 | 18,452 | 58.18 | 17.34 | 30.21 | 57.4 | 12.095 | 9,121 | .494 | 2.023 |
| 35 | 6 | 30 | 30.27 | 1,631 | 74.1 | 74.5 | 81.7 | 46 | 2.140 | 9.483 | 11.563 | 9.155 | .07380 | 12,206 | 0.592 | 7,226 | 44 55 | 13.15 | 24.22 | 54.3 | 13.099 | 9,548 | .719 | .757 |
| 36 | 5 | 30 | 30.23 | 1,631 | 73.1 | 72.6 | 78.4 | 40 | .032 | 7.003 | 7.934 | 7.304 | .07419 | 10,876 | 1.301 | 14,150 | 57 62 | 17 66 | 31.33 | 56.4 | 13.173 | 9,548 | .600 | 1.482 |
| 37 | 5 | 30 | 30.27 | 1,631 | 73.6 | 73.3 | 77.7 | 39 | .000 | 6.534 | 6.534 | 5.544 | .07439 | 9,461 | 2.040 | 19,300 | 63.79 | 19.84 | 34.70 | 57.2 | 13.231 | 9,554 | .494 | 2.020 |
| 38 | 4 | 30 | 30.00 | 1,631 | 72.0 | 71.3 | 75.4 | 45 | .000 | 5.039 | 5.039 | 3.265 | .07403 | 7,280 | 3.382 | 24,620 | 75.51 | 19.52 | 41.09 | 47.5 | 13.145 | 9,548 | .383 | 2.579 |

REPORT OF TEST OF 30-INCH KEITH FAN.

Item. *Method of Obtaining Items in Tables I and II.*

1, 2, 3. For reference.

4. From data corrected for temperature according to Smithsonian Meteorological Tables.

5. From direct-reading revolution counter.

6, 7, 8. Observed.

9. From observations of wet and dry-bulb thermometer.

10. From observations of water gage connected to impact tube at center of suction duct.

11. Average of readings of three draft gages connected to static tubes in test compartment.

12. Item 10 + item 11.

13. From horizontal and vertical tube traverses of discharge nozzle.

14. Weight of one cubic foot of air corresponding to conditions of items 4, 8 and 9.

15. From formula:

$$V = 1096\sqrt{\frac{h}{d}};$$

Where $h$ = velocity head = item 13.

$d$ = density of air = item 14.

16. Area of particular nozzle in use during test.

17. Item 15 × item 16.

18. Net load on beam of electric dynamometer.

19. From formula

$$\frac{pv}{33{,}000} = \text{A.H.P.}$$

Where $p$ = air pressure in pounds per square foot = 5.19 × item 12.

$v$ = volume in cubic feet = item 17.

20. From formula

$$\text{H.P.} = \frac{FN}{3{,}000}.$$

Where F = net brake load = item 18.

N = item 5.

$\frac{1}{3{,}000}$ = constant for electric dynamometer.

21. $\frac{\text{Item 19.}}{\text{Item 20.}}$

22. Water pressure corresponding to tip speed.

$$\text{T.S.P.} = \left(\frac{\pi BN}{720}\right)^2 \times \frac{1}{2g} \times \frac{d \times 12}{62.3}.$$

B = mean diameter over tips of blades = 34.22 inches.
N = r.p.m. = item 5.
$g$ = acceleration due to gravity = 32.2.
$d$ = density of air = item 14.

23. Volume of cylinder 34.22 inches diameter, 11 inches high × item 5.
24. $\frac{\text{Item 11}}{\text{Item 22}}$
25. $\frac{\text{Item 17}}{\text{Item 23}}$

16. Plot No. 3 compares at constant speed of 1,631 r.p.m. the performance of the fan with and without diffuser. The head for the same discharge is from 0.75 to 1.42 inches greater with diffuser than without and the efficiency from 7.5 per cent. to 12.1 per cent. greater. The efficiency characteristic without diffuser is a flatter curve than the lower, insuring more economical operation over a wide range than without diffuser.

17. For a given blast area the discharge will vary as the speed, the head as the square of the speed, and the air and shaft horsepowers as the cube of the speed. Such characteristics are shown in the curves on Inclosure J for second largest nozzle and fan without diffuser, tests 26, 30, 34 and 37.

18. The disturbances in the suction duct are very violent at low discharges, being only slightly less with diffuser than without. When discharging 5,726 cubic feet of air against a pressure of 6 inches water gage the suction measured by an impact tube one foot from the fan inlet was 1.630 inches at 1,394 r.p.m. without diffuser and 1.316 inches at 1,296 r.p.m. with diffuser. A slip of paper released at the inlet would be whirled around the inside of the trunk often traveling back and forth in the ten-foot length of trunk from the funnel to the inlet before finally being carried through the fan and out the discharge. In calculating overall mechanical efficiency the fan was credited with the air work expended on the suction side, although there is no useful result therefrom on the discharge side. The effect is appreciable only in the first two tests of each group. Vibration and noise were excessive in runs 3, 11, 14, 27, 35, 36 and 37. All runs at 1,631 r.p.m.

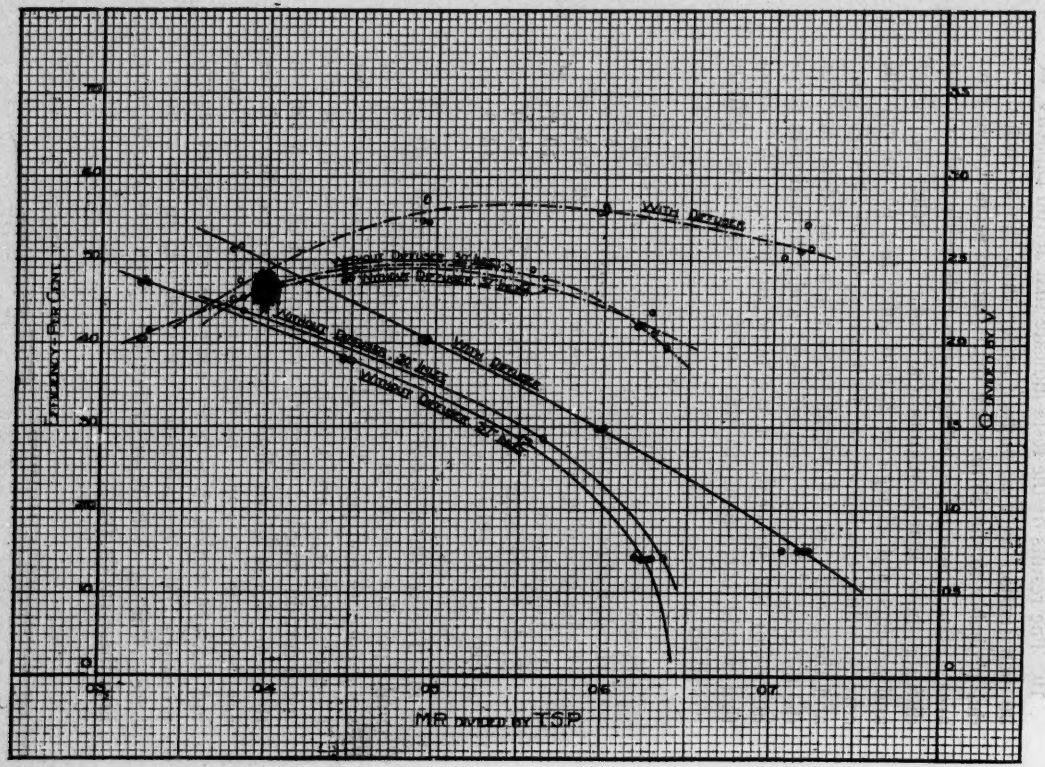
gave more noise than desirable for the operation of two blowers in one air-tight compartment.

## VI.—SUMMARY AND CONCLUSIONS.

19. The results of the tests indicate that at full capacity the 30-inch suction duct is slightly better than the 37-inch duct, due probably to the elimination of a sudden reduction in flow area. The comparison between the tests with and without diffuser shows that to maintain a head of 5 inches of water and produce a discharge of 25,000 cubic feet per minute the corresponding efficiencies are 46.9 per cent. and 39.8 per cent. and the speeds are 1,638 and 1,785 r.p.m., an advantage for the diffuser of 7.1 per cent. efficiency and a reduction in revolutions of 8.2 per cent. for the same air work. The rating that would correspond to this performance with diffuser would be:

| | |
|---|---|
| Speed, r.p.m. | 1,638 |
| Head, inches of water | 5 |
| Discharge, cubic feet per minute | 25,000 |
| Air horsepower | 19.7 |
| Brake horsepower | 42.0 |
| Mechanical efficiency, per cent | 46.9 |

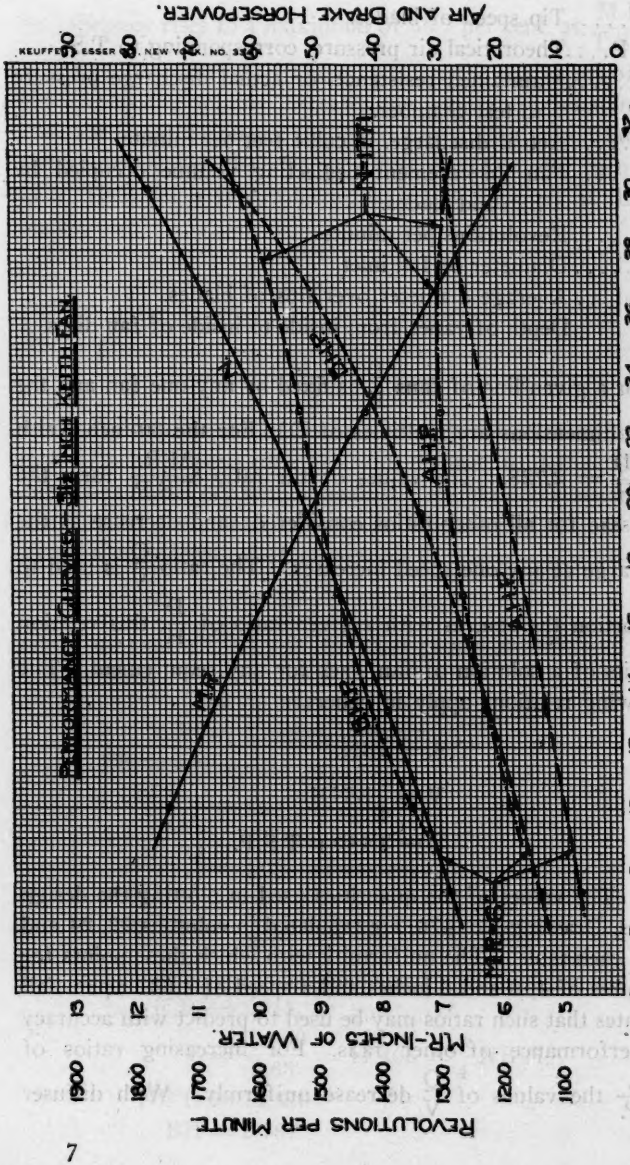
20. The results deduced from the tests furnish data which make possible the design of geometrically similar fans for the same purpose and permit of plotting their characteristics for all speeds and discharges. The method assumes that at any speed certain ratios of pressure, quantity and efficiency remain the same for a given discharge area. The validity of this assumption is established by reference to the curves on Drawing No. 476, which show how nearly the test points at several pressures and speeds register when reduced to the common base. The symbols representing the quantities which enter into these ratios are:




KEY.

M.R. Maintained resistance.
T.S.P. Tip-speed pressure.
Q. Actual discharge.
V. Fan displacement.
D divided by H=2.728.
B divided by H=3.110.
Full lines indicate Q÷V.
Broken lines indicate efficiency.

DESIGN CURVES, 30-INCH KEITH FAN.—INCLOSURE (K).


Performance Curves—30 Inch Keith Fan
Revolutions per Minute.
M.R.-Inches of Water.
Thousands of Cubic Feet per Minute
Air and Drake Horsepower.
KEUFFEL & ESSER CO., NEW YORK. NO. 333.

INCLOSURE (L).

DESIGN CURVES, 30-INCH KEITH FAN.—INCLOSURE (K).

T.S.....Tip speed of blades.
T.S.P....Theoretical air pressure, corresponding to T.S.
M.R.....Maintained resistance, or actual discharge pressure in test chamber.
Q.....Actual discharge, in cubic feet per minute.
V.....Fan displacement neglecting volume occupied by hub and blades.
N.....Revolutions per minute.
D.....Diameter of fan inlet.
B.....Average diameter over tips of blades.
H.....Depth of vanes from inside to inside of fan discs.

For a given line of fans $\frac{D}{H}$ is fixed by any one fan and the other dimensions are multiples of D. For the 30-inch Keith fan $\frac{D}{H} = 2.728$. The values of the ratio $\frac{M.R.}{T.S.P.}$ form the abscissae for all other ratios considered, and decrease as the discharge at any one speed increases. The ratio $\frac{Q}{V}$ is used to proportion the fan, as with a given ratio of $\frac{D}{H}$, this relation permits of a solution of fan sizes for any requirement. The following relations apply in general:

$$T.S. = \pi BN.$$

$$V = \frac{\pi}{4} B^2 NH.$$

$$T.S. = 4{,}050 \sqrt{T.S.P.}$$

21. The values of the quantities used in plotting the design curves, Drawing No. 476, are tabulated in columns 22, 24 and 25 of each table. The uniformity with which these points for pressures of 4, 5 and 6 inches and a speed of 1,631 r.p.m. fall indicates that such ratios may be used to predict with accuracy the performance of other fans. For increasing ratios of $\frac{M.R.}{T.S.P.}$ the values of $\frac{Q}{V}$ decrease uniformly. With diffuser

the efficiency rises to a maximum of 56.7 per cent. at $\frac{\text{M.R.}}{\text{T.S.P.}} = .545$ and then falls off. These facts are the equivalent of saying that as the volume increases the maintained resistance at constant speed decreases, the speed to maintain a constant resistance increases, the efficiency first increases and then falls off. Then for a fan to have maximum efficiency at full capacity,

$$\frac{\text{M.R.}}{\text{T.S.P.}} = .545,$$

and for a fan to attain maximum efficiency before full rated discharge

$$\frac{\text{M.R.}}{\text{T.S.P.}} < .545.$$

22. As an example of the use of such curves, take the design of a high-speed cone fan with diffuser to deliver 30,000 cubic feet of air per minute against a maintained resistance of 6 inches of water, and to have an efficiency of 45 per cent. From a consideration of paragraph 21

$$\frac{\text{M.R.}}{\text{T.S.P.}} < .545,$$

since full-load efficiency is less than maximum efficiency.

Therefore

$$Q = 30{,}000 \text{ cubic feet per minute.}$$

$$\text{M.R.} = 6 \text{ inches of water.}$$

$$\text{Efficiency} = .45.$$

$$\frac{\text{M.R.}}{\text{T.S.P.}} = .367 \quad \text{(from curve).}$$

$$\frac{Q}{V} = 2.63 \quad \text{(from curve).}$$

$$\text{T.S.P.} = \frac{6}{.367} = 16.35.$$

$$\text{T.S.} = 4{,}050\sqrt{16.35} = 16{,}390 \text{ feet per minute.}$$

$$\text{N.B.} = 5{,}215.$$

$$V = \frac{30{,}000}{2.63} = 11{,}405 = \frac{\pi}{4}\text{NB}^2\text{H}.$$

$$\text{NB}^2\text{H} = 14{,}520.$$

$$\text{BH} = 2.787.$$

$\frac{B}{H} = 3.110.$

$H = .946$ feet $= 11.36$ inches. Use $11\frac{3}{8}$ inches.

$B = 3.110 \times 11.36 = 35.32$ inches.

Corresponding tip diameters of blades are, minimum, $33\frac{1}{2}$ inches; maximum, $37\frac{1}{8}$ inches.

$D = 2.728 \times 11.36 = 31.10$ inches. Use $31\frac{1}{2}$ inches.

$N = 1{,}771$ r.p.m.

$$\text{Air horsepower} = \frac{5.19 \times 6 \times 30{,}000}{33{,}000} = 28.3.$$

$$\text{Brake H.P.} = \frac{28.3}{.45} = 62.9.$$

Similarly the characteristics at other rates of discharge would be given by calculations based on other ratios from the curves as shown in accompanying Table III. Speed is constant for first four points and maintained resistance constant for last four points. The performance is given by the curves of Inclosure L.

TABLE III.

| Number. | $\frac{\text{M.R.}}{\text{T.S.P.}}$ | $\frac{Q}{\sqrt{V}}$ | M.R. inches of water. | Revolutions per minute. | Cubic feet per minute. | Air H.P. | Mechanical efficiency, per cent. | Brake H.P. |
|---|---|---|---|---|---|---|---|---|
| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
| 1 | .700 | .88 | 11.44 | 1,771 | 10,040 | 18.1 | 52.1 | 34.7 |
| 2 | .600 | 1.48 | 9.81 | 1,771 | 16,880 | 26.0 | 56.0 | 46.5 |
| 3 | .500 | 2.00 | 8.18 | 1,771 | 22,810 | 29.5 | 56.0 | 52.6 |
| 4 | .367 | 2.63 | 6.00 | 1,771 | 30,000 | 28.3 | 45.0 | 62.9 |
| 5 | .700 | .88 | 6.00 | 1,283 | 7,280 | 6.9 | 52.1 | 13.2 |
| 6 | .600 | 1.48 | 6.00 | 1,385 | 13,200 | 12.5 | 56.0 | 22.3 |
| 7 | .500 | 2.00 | 6.00 | 1,516 | 19,520 | 18.4 | 56.0 | 32.9 |
| 8 | .367 | 2.63 | 6.00 | 1,771 | 30,000 | 28.3 | 45.0 | 62.9 |

23. The design as per above figures works out for a required speed of 1,771 r.p.m. to give rated conditions. While the fan would be safe to operate at that speed, the noise might

be excessive, so that a higher value should have been assumed for the mechanical efficiency, at rated capacity, which would result in a wheel of greater diameter and lower speed to do the same work.

## VII.—RECOMMENDATION.

24. Except for noise the Keith fan has shown itself as good as any heretofore tested at this station and its use in the naval service with suction trunk of the same diameter as fan inlet and with diffuser is recommended. The efficiency curve remains flat over a wide range of operation, so that the economy of the unit at low duty is practically the same as at forced-draft conditions except as effected by the driving engine.

# NOTES ON ENGINEERING INSPECTION WORK.

By Robt. Crawford, Chief Engineer, U. S. N., Retired.

Note.—The accompanying notes have been prepared with the idea of giving some general elementary ideas and points concerning the inspection work carried on by the force under the Inspector of Machinery for vessels building for the Navy.

As this work has an intimate relation to the engineering matters on a vessel after commissioning, it is thought that it will be of special interest to those officers doing engineering duty who may have in contemplation a tour of duty at Engineering Inspection work. It is not intended to be a comprehensive discussion of the whole field, but only treats of certain matters.

A brief outline of what the Inspection force consists and the general division of duties will first be given; notes on the details of the work following.—H. C. D.

## ORGANIZATION OF INSPECTION FORCE.

The organization of the inspection force and their detailed duties and the system of office procedure is left to the Inspector in Charge, subject to the various rules and regulations issued by the Navy Department. Inspectors are expected to maintain their office and its personnel in the most efficient condition possible, and the responsibility for securing the best results from the force on hand will rest with the Inspector.

There will usually be included the following:

Assistant Inspectors of Machinery, one of whom is assigned to the Electrical Inspection Work.

Drafting force.—Engineering and electrical draftsmen.

Clerical force.—Clerks, stenographers and messengers.

Weight-compiling force.—Clerks, draftsmen or special mechanics.

Outside inspection force.—Commissioned and warrant officers, special mechanics, draftsmen, enlisted petty officers assigned to duty in connection with vessels building.

## ASSISTANT INSPECTORS.

Officers detailed for duty as Assistant Inspectors will make themselves thoroughly familiar with all work assigned to them. They will keep the Inspector of Machinery informed of all matters concerning the progress of work, inspections, desirable changes or alterations, unsatisfactory work, etc. They will be prepared to furnish the Inspector of Machinery with complete information on any subject concerning their work.

They will prepare such letters as may be required in answer to correspondence that may be referred to them.

They shall perform such work of inspection as will keep them thoroughly cognizant of the progress of the work of manufacture and installation, and will assist in the criticism of all plans and drawings. They shall see that all matters coming under their cognizance are in compliance with the approved instructions, and will endeavor to gain as much information in regard to the methods of work and plans as may be of service to them.

Where several assistant inspectors are provided, one of them will be specially assigned to inspection of electrical work.

Assistant inspectors are to a large extent assigned with the idea of giving officers in the lower grades an opportunity for becoming familiar with engineering practice, and especially with the special features of engineering inspection. They will therefore be required to inform themselves on the general engineering problems involved in the building of naval machinery, engineering shop practice and the general matters of in-

spection work, so as to fit themselves for future responsible engineering duty.

### ELECTRICAL INSPECTOR.

When sufficient assistant inspectors are detailed one will be assigned in charge of the inspection work of the electric plant and installation. Electrical draftsmen or aids will be placed directly under his supervision. While the force charged with the inspection of electrical installation is in a measure apart, nothing is to be construed to preclude the force from doing any other inspection work whenever occasion arises to make such a course desirable.

### WARRANT OFFICERS.

Warrant officers who are ordered to duty as assistants to the Inspector of Machinery will in general perform duty in connection with the outside inspection work. They shall also perform such additional duty as may be required of them by the Inspector of Machinery.

The general requirements for the inspection force and specific instructions concerning the work carried on by the force under the Inspector of Machinery are given in the Bureau of Steam Engineering's "General Instructions to Inspectors of Machinery." These instructions cover procedure for carrying on the inspection work, manner of submitting correspondence, drawings, preparations of reports, certificates, etc. The following notes are intended to cover details which come up in connection with such work with which the inspection force, especially the outside inspection force, is connected.

### INSPECTION.

The study of and reference to the machinery specifications and drawings is a first requisite in effective inspection work. The drawings should be studied as they are received from the

Bureau with its approval, since by this means the inspector is informed in advance of the inspections he may be called upon to make in matters of design and finish.

While the hydrostatic tests and inspection of pipes is a comparatively simple operation, the tests and inspections of valves for the various pipe lines must be conducted with much care and close observation. Castings for valve bodies, of whatever material made, are liable to certain characteristic defects. Steel castings, for example, are rarely free from one or more, and sometimes possess all, of the following defects: porosity, blow holes, sand pits, blisters and cracks, each of which may be remedied if not of too serious a nature. If the porosity is local and slight, as indicated by the manner in which the water oozes through the casting, it may be satisfactorily remedied by peening, but if at all general the casting should be condemned. Should the leak indicate a small hole, this may be safely corrected by plugging with a screw plug with the end slightly riveted over. The plug should not be of a diameter greater than the thickness of the casting. Of all the defects to which these castings are liable, cracks caused by shrinkage of the material in cooling are most objectionable and difficult to correct. They may readily be detected, however, and their extent determined by a continuous line of moisture oozing through the casting, which if wiped off will quickly reappear. Very small cracks may be satisfactorily caulked, while more serious ones may be corrected by "burning in" new metal by one of several well known processes which, if well done, entirely removes the defect. When this has been done the valve casting must again be subjected to the maximum water test, and, if tight, may be accepted. In cases where the cracks are much extended no attempt to repair the defect will be authorized, and the casting must be condemned. Steel castings are particularly liable to blisters and sand pits, which are easily dealt with; the latter by cleaning out the sand until good metal is reached; the former by chipping off the blisters to determine their thickness, which is usually negligible.

Brass valve castings are particularly liable to a form of porosity which becomes apparent when the casting is subjected to hydrostatic pressure, the water oozing out through surfaces that have been machined. The cause for this is not well understood. It would appear, however, to be due to infinitely small spaces formed in the metal during the process of cooling after casting. This porosity may sometimes be remedied by subjecting the casting to a pressure of steam of from 50 to 100 pounds for a number of hours. When, however, the defect is found not to yield to this treatment, or to light peening, the casting must be condemned.

Iron castings for valve bodies will also often be found to contain porous spots, particularly in angles formed with the body by flanges, brackets and other extensions, frequently necessitating condemnation. These castings are also liable to spongy spots, difficult to remedy. Where these exist to an extent necessitating treatment other than slight peening, it is advisable to have the casting replaced.

*Shop and Ship Inspections.*—It is not practicable to lay down definite rules for the guidance of the inspection force in the supervision of the construction and installation of machinery for the reason that conditions, as well as requirements, are constantly varying. It may, however, be stated that constant vigilance in the supervision of all work passing through the various stages of construction and installation is essential to the proper performance of these duties.

The greater number of the shop inspections consist of hydrostatic tests of sections of piping for the various systems of steam, water and other pipe lines for the vessel under construction, and the various sizes and kinds of valves belonging to them. For each part tested the maximum test pressure will be given on the drawing from which the part was made, or in the printed specifications, both of which should prescribe the test pressure for the various classes of pipes and valves. If the part tested is found free from leaks and the workmanship in its construction is satisfactory and completed, the part

is stamped with the letter " T," indicating that it has received the required structural test, and if satisfactory in this respect and in workmanship it is further stamped " OK," which gives authority for the part to be taken on board ship and installed. Should the test reveal structural defects or the workmanship be found defective, the part is not stamped as indicated until it has had the defects causing its rejection removed. Should the defects be of such a character as to render the part undesirable for service, it is stamped " NG," indicating that it has been rejected.

When the hydrostatic test has proven the tightness of the valve body, the tightness of the valve disc in its seat must be determined. This test must be conducted with considerable care to avoid injury to the disc or its seat.

Where the test pressure is applied to the back of the valve disc, as is usual and desirable, and the seat has been ground, there will be little difficulty in securing absolute freedom from leak. If, however, the test pressure must be applied to the face or under side of the disc, to conform with the position of the valve when in service, much greater difficulty may be experienced in securing tightness, though the construction and workmanship be in all respects satisfactory. This is due to the disc springing under the severe hydrostatic pressure upon its face necessary in the test. A good procedure in conducting this test is, first to close the valve (disc) lightly by hand and increase the hydrostatic pressure until the seat leaks, which is likely to occur while the pressure is well below the test limit; again set the valve down by hand until the leak stops and increase the hydrostatic pressure. By continuing this tentative process until the maximum test pressure has been brought upon the face of the valve, undue straining and injury to the valve seat is avoided, and unless the valve and seat have been badly ground together they will be found to be tight.

Should, however, the seat continue to leak slightly, the force in seating the valve may be increased until the leak stops and

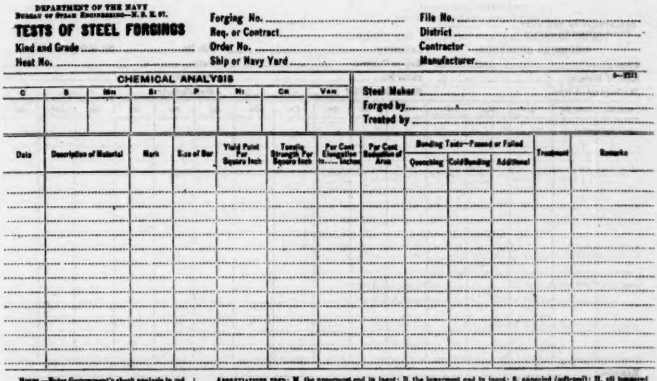
without injury to the seat or valve, due to the fact that the force thus applied will be almost counterbalanced by the hydrostatic pressure upon the valve face.

The seats of valves are often seriously injured by jamming the valves down hard before applying the hydrostatic pressure. No more force should be exerted at any time in seating a disc valve than that necessary to make it tight.

Steam valves are also required to be tested for tightness under steam pressure.

*Machine Shop Inspections.*—These inspections are mainly connected with two principal shop operations designated as *machining* and *fitting up;* the one including all work done by machinery, as the term implies, the other all forms of hand work not otherwise designated. There are in addition to these inspections, physical tests of materials which the inspector must witness, that he may be able to certify that they possess the strength and other essential characteristics called for in the Bureau's machinery specifications. Also, there must be hydrostatic tests of valves and many other appliances for the general machinery which when in use will be required to sustain internal pressure. Notice will be sent to the inspector when parts are ready to be tested, and when the test is completed, the results being satisfactory, the parts will be marked, as elsewhere described, with the letter " T." If all work on the part has been satisfactorily completed, it will be additionally stamped " OK" and be regarded as ready for installation. All such tests are recorded on approved forms or in a book for future reference, the name or initial of the inspector appearing as a part of the record.

The notice for a physical test will be accompanied by samples of the metal containing identification marks and known as test pieces. These are finished in form and size as prescribed in the specifications for the kind of tests required. Monthly reports of all such tests are sent the Bureau of Steam Engineering. The following are sample forms for reporting these tests:



DEPARTMENT OF THE NAVY
BUREAU OF STEAM ENGINEERING—N. A. E. W.

**TESTS OF STEEL FORGINGS**

Kind and Grade .....
Heat No. .....

Forging No. .....
Req. or Contract .....
Order No. .....
Ship or Navy Yard .....

File No. .....
District .....
Contractor .....
Manufacturer .....

CHEMICAL ANALYSIS

| C | S | Mn | Si | P | Ni | Cr | Van |
|---|---|---|---|---|---|---|---|
| | | | | | | | |

Steel Maker .....
Forged by .....
Treated by .....

| Date | Description of Material | Mark | Size of Bar | Yield Point Per Square Inch | Tensile Strength Per Square Inch | Per Cent Elongation in ..... Inches | Per Cent Reduction of Area | Bending Tests—Passed or Failed: Quenching | Cold Bending | Additional | Treatment | Remarks |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
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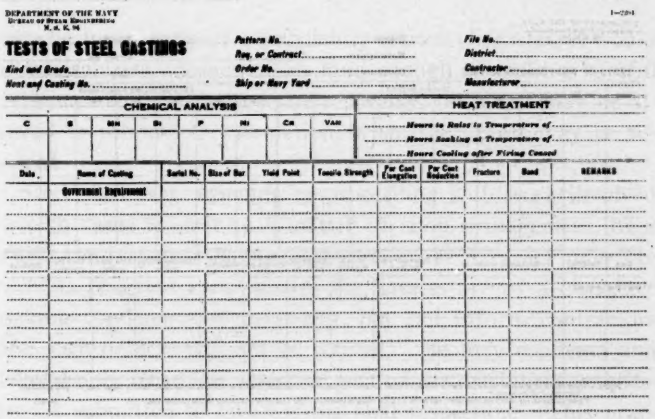
Norm.—Enter Government's check analysis in red ink. Government requirements, physical and chemical, should be given as well as actual results.

ABBREVIATIONS USED: M, the requirement end in ingot; B, the lowermost end in ingot; S, annealed (soft-end); H, oil tempered (hardened); —L, transverse; B, B₁, B₂, etc., number of slab from bottom of ingot; F₁, F₂, F₃, etc., number of objects manufactured in one; I, at one-half radius (inner); O, at circumference (outer); CP, near crank pin; C, I, from block removed to form crank.



DEPARTMENT OF THE NAVY
BUREAU OF STEAM ENGINEERING
N. A. E. W.

**TESTS OF STEEL CASTINGS**

Kind and Grade .....
Heat and Casting No. .....

Pattern No. .....
Req. or Contract .....
Order No. .....
Ship or Navy Yard .....

File No. .....
District .....
Contractor .....
Manufacturer .....

CHEMICAL ANALYSIS

| C | S | Mn | Si | P | Ni | Cr | Van |
|---|---|---|---|---|---|---|---|
| | | | | | | | |

HEAT TREATMENT

.....Hours to Raise to Temperature of.....
.....Hours Soaking at Temperature of.....
.....Hours Cooling after Firing Ceased

| Date | Name of Casting | Serial No. | Size of Bar | Yield Point | Tensile Strength | Per Cent Elongation | Per Cent Reduction | Fracture | Bend | REMARKS |
|---|---|---|---|---|---|---|---|---|---|---|
| | Government Requirement | | | | | | | | | |
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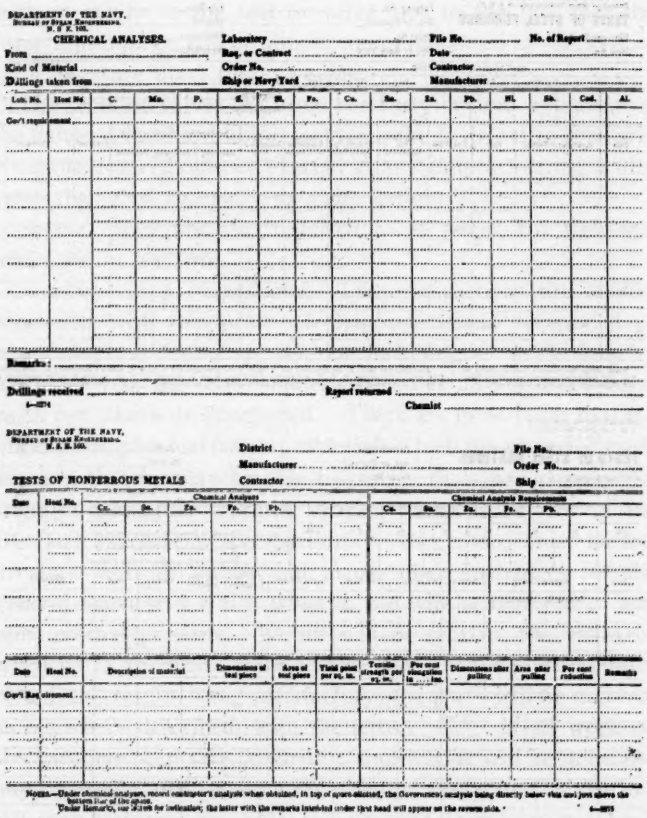
Chemical analysis of the materials used is required by the specifications. The contractors furnish the analysis obtained by their own chemist, and also supply samples of the metal in the form of clean filings or borings to the inspector, to be sent to the nearest Government chemist. These two analyses are submitted to the inspector for approval, and the report is forwarded to the Bureau of Steam Engineering.



DEPARTMENT OF THE NAVY,
Bureau of Steam Engineering.
S. E. E. 105.

CHEMICAL ANALYSES.

Laboratory ..... File No. ..... No. of Report .....
From ..... Req. or Contract ..... Date .....
Kind of Material ..... Order No. ..... Contractor .....
Drillings taken from ..... Ship or Navy Yard ..... Manufacturer .....

| Lab. No. | Heat No. | C. | Mn. | P. | S. | Si. | Fe. | Cu. | Sn. | Zn. | Pb. | Ni. | Sb. | Cod. | Al. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Gov't requirement | | | | | | | | | | | | | | | |
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Remarks: .....

Drillings received ..... Report returned .....
4—8274 Chemist .....

DEPARTMENT OF THE NAVY,
Bureau of Steam Engineering.
S. E. E. 105.

TESTS OF NONFERROUS METALS.

District ..... File No. .....
Manufacturer ..... Order No. .....
Contractor ..... Ship .....

| Date | Heat No. | Chemical Analysis | | | | | | | Chemical Analysis Requirements | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Cu. | Sn. | Zn. | Pb. | Ni. | | | Cu. | Sn. | Zn. | Pb. | Ni. | | |
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| Date | Heat No. | Description of material | Dimensions of test piece | Area of test piece | Yield point per sq. in. | Tensile strength per sq. in. | Per cent elongation in ... in. | Dimensions after pulling | Area after pulling | Per cent reduction | Remarks |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Gov't Requirement | | | | | | | | | | | |
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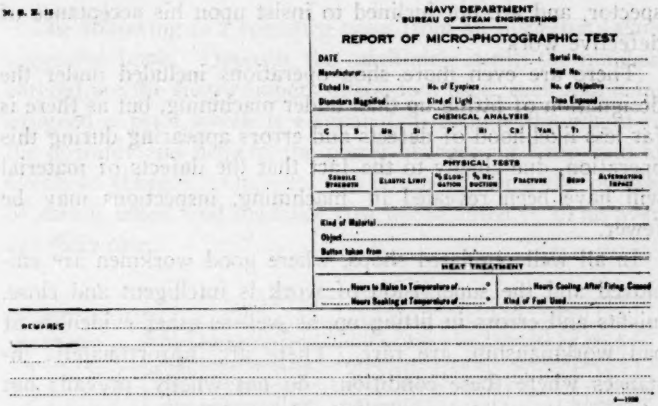
Notes.—Under chemical analyses, record contractor's analysis when obtained, in top of space allotted, the Government analysis being directly below; this and just above the bottom line of the space.
Under Remarks, use figure for indication; the latter with the remarks intended under that head will appear on the reverse side. 4—8275

Micro-photographic tests also are sometimes made. The accompanying form being used.

While machining and fitting up parts of the machinery is in progress, inspection of the various operations under these designations will require to be practically continuous. Especially is this true of machining, since it is during this operation that defects in material are most likely to be revealed and should be carefully examined at once. Should these de-



N. S. E. 15

NAVY DEPARTMENT
BUREAU OF STEAM ENGINEERING

REPORT OF MICRO-PHOTOGRAPHIC TEST

DATE ..... Serial No. .....
Manufacturer ..... Heat No. .....
Etched In ..... No. of Eyepiece ..... No. of Objective .....
Diameters Magnified ..... Kind of Light ..... Time Exposed .....

CHEMICAL ANALYSIS

| C | S | Mn | Si | P | Ni | Cr | Van. | Ti | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | | | |

PHYSICAL TESTS

| Tensile Strength | Elastic Limit | % Elongation | % Reduction | Fracture | Bend | Alternating Impact |
|---|---|---|---|---|---|---|
| | | | | | | |

Kind of Material .....
Object .....
Button taken from .....

HEAT TREATMENT

..... Hours to Raise to Temperature of ..... ° ..... Hours Cooling After Firing Ceased
..... Hours Soaking at Temperature of ..... ° Kind of Fuel Used .....

REMARKS .....

fects be found to be of a sufficiently serious character, it would be advisable to suspend work long enough to make a careful examination. If found to be reparable, the work may be allowed to proceed to completion, when the defect may be repaired.

Since, however, stopping or proceeding with work involved in such cases is entirely a matter of shop management, over which the inspector has no jurisdiction, he may not interfere with its progress even though the defects are of a character making condemnation necessary, but can inform contractors that such defects will not be waived. He may condemn and discard only when the defective part of the machinery in question is submitted to him for his final inspection. This practice is reasonable, as it affords the contractors proper opportunity to repair defects, whether of a light or serious nature, which they are often capable of satisfactorily doing. If at any time the contractors will not accept the inspector's decision as to the fitness for use of a part of the machinery in course of construction found to be imperfect, the matter can be referred to the Bureau for final decision. It may be stated that experience in handling many cases of this kind has shown that the contractors are usually disposed to act with the in-

spector, and are not inclined to insist upon his acceptance of defective work.

There are even more shop operations included under the designation of fitting up than under machining, but as there is far less likelihood of defects and errors appearing during this operation, due chiefly to the fact that the defects of material will have been revealed in machining, inspections may be fewer.

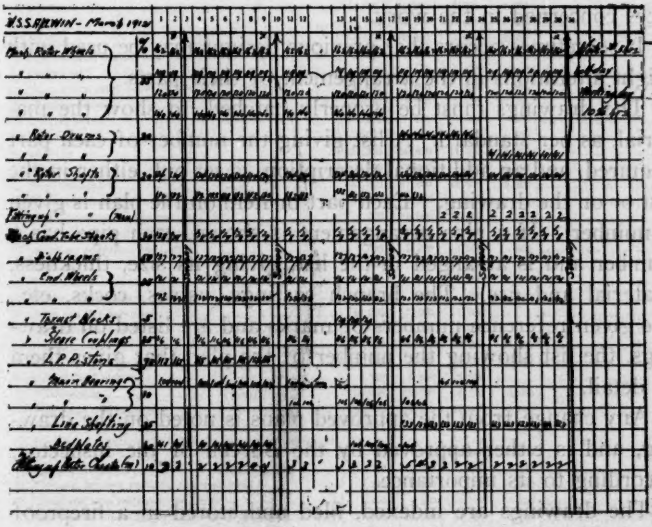
In all well regulated shops, where good workmen are employed and the supervision of work is intelligent and close, misfits and errors in fitting up, as well as other evidences of bad workmanship, are rare. There are, unfortunately, instances where these conditions do not wholly prevail; but whatever the conditions, it is the duty of the inspection force to keep in close touch with all work in progress, since by so doing not only is better work insured, better service given, but the inspector's knowledge of methods of doing work is improved.

Daily inspections of work in process of machining and fitting up are entered in an inspector's book conveniently arranged for such records. In the machining operations record is made of the machines, each of which is designated by its shop number, and the number of hours it is in use, and its per hour valuation.

In entering the fitting-up operations the number of men employed on each operation, with the average daily wage and number of hours of daily work, are noted.

This daily inspection record is of especial importance to the inspector, as it affords a reliable basis on which to check the labor earnings of the contractors when payment on a contract is asked for. The several money allotments for groups of machinery appearing in Form 102 are only approximately correct, and the amount earned under each item should be carefully checked when sending in monthly reports and reports accompanying applications for payments, by taking the shop record, together with the record of material received and accepted by the inspector.

The following is a specimen page from an inspector's daily inspection book. Opposite the machining items, the numbers entered are the shop numbers of the tool used, each tool being assigned a price which is estimated to include the wages of the operator and helpers, the power charges, up keep, and all overhead charges. Opposite the fitting-up items the number of men is noted, and the labor cost is computed from an average daily rate.



SPECIMEN PAGE.—INSPECTOR'S DAILY INSPECTION BOOK.

## DRAWINGS.

The drawings for any part of the machinery are made in pencil on drawing paper, and tracings are made from them; or if a sketch of some minor part is desirable, it is made on thin paper and inked in, so as to obtain a print directly from it.

The object to be drawn is sketched by the designing engineer, whose idea is conveyed to the chief draftsman to be elaborated into a drawing. This drawing is made in pencil to convenient scale to get it on a sheet 27 inches × 40 inches,

or a multiple of that size, the size being a Bureau requirement. From the pencil drawing a tracing is made. The tracing, or a blue print therefrom, is submitted to the Bureau for approval, through the office of the Inspector of Machinery, who examines it closely to see that it conforms in all respects to the specifications and the practice established by the Bureau.

When the tracing or blue print is returned with the Bureau's approval, blue or other prints are made from the tracing, by the solar or electric process, for the use of the shops and offices concerned in the work.

It sometimes occurs that previous designs may be used with slight alterations in the arrangement of their parts.

The drawings must be properly hatched, to show the material, as per standard. A list giving the number of each part required, and an additional description, is set out, either in the list or on the drawing. Each part or item on the plan is given a number. If a pipe arrangement is shown, each pipe has a number and is arranged in the list, giving its size, thickness, material and use. The branch castings, valves, cocks, etc., are given reference numbers or marks and are listed on drawings, the list showing the number of the plan that gives them in detail.

Any change from the approved plans is noted on the drawing, and is either approved by the Bureau or the inspector, according to its importance.

The drawings are indexed, filed and stored in a fireproof safe, and a careful record is kept of them when issued to the different shops where the work is to be done.

### DEGREE OF COMPLETION, PAYMENTS.

The contract provides that payment for the vessel shall be made in a specified number of equal instalments as the work progresses; provision being made for the reservation of several payments pending the preliminary acceptance of the ship. In the case of a battleship the payments are in fifty (50) instalments, with three (3) reserved. In the case of a destroyer, in twenty (20) payments, with two (2) reserved.

NAVY DEPARTMENT.
BUREAU OF STEAM ENGINEERING.
E. E. K. 326.

## OFFICE OF THE INSPECTOR OF MACHINERY
## AT BATH IRON WORKS, BATH, ME.

**Report on Machinery of Torpedo Boat Destroyer No. 60, Showing Condition of Work on .....**

| | 1 | 2 | 3 | 4 | 5 | 6 |
|---|---|---|---|---|---|---|
| | WEIGHT (pounds.) | ALLOTMENT. | | | MATERIAL RECEIVED. | LABOR COMPLETED. |
| | | Total. | Material. | Labor. | | |
| 1. General superintendence, office expenses, freight, etc. | | $92,000 | | $92,000 | | |
| 2. Drafting and patterns | | 48,000 | | 48,000 | | |
| 3. Trial trip expenses | | 6,000 | $3,000 | 3,000 | | |
| 4. Turbine royalties | | 8,750 | 8,750 | | | |
| 5. Erecting engines | | 2,100 | | 2,100 | | |
| 6. Cylinders | 36,000 | 10,000 | 6,000 | 4,000 | | |
| 7. Spindles | 43,000 | 42,000 | 38,000 | 4,000 | | |
| 8. Blading | 8,650 | 6,800 | 5,600 | 3,200 | | |
| 9. Brasses, main bearing and thrust | 2,800 | 1,800 | 1,100 | 700 | | |
| 10. Oiling gear | 1,500 | 800 | 400 | 300 | | |
| 12. Drain cocks, relief valves, throttle valves, and gear | 2,800 | 1,600 | 700 | 900 | | |
| 13. Receiver pipers | 7,000 | 1,850 | 1,100 | 750 | | |
| 14. Turning gear | 200 | 150 | 80 | 100 | | |
| 15. Propellers | 8,000 | 5,000 | 3,700 | 1,300 | | |
| 16. Shafting | 39,800 | 20,800 | 17,500 | 2,800 | | |
| 17. Bearings | 3,300 | 2,200 | 1,100 | 1,100 | | |
| 18. Main condensers | 42,000 | 17,900 | 12,500 | 5,400 | | |
| 19. Main air pumps | 11,000 | 9,500 | 9,500 | | | |
| 20. Main circulating pumps and engines | 4,000 | 5,000 | 5,000 | | | |
| 21. Main injection and outboard delivery valves and piping | 5,500 | 2,900 | 1,900 | 1,000 | | |
| 22. Auxiliary condenser and pumps | 6,000 | 2,200 | 1,800 | 400 | | |
| 23. Feed tank | 1,900 | 800 | 150 | 240 | | |
| 24. Feed heater | 3,600 | 1,900 | 600 | 1,300 | | |
| 25. Waste, oil, and tallow tanks | 1,800 | 400 | 100 | 300 | | |
| 26. Main and auxiliary feed pumps | 10,000 | 7,000 | 7,000 | | | |
| 27. Miscellaneous pumps | 7,000 | 7,000 | 7,000 | | | |
| 28. Boilers | 272,000 | 88,000 | 42,000 | 46,000 | | |
| 29. Uptakes and smoke pipes | 27,000 | 4,200 | 1,700 | 2,500 | | |
| 30. Forced draft blowers | 8,500 | 6,450 | 6,450 | | | |
| 31. Water piping | 24,000 | 11,800 | 8,100 | 3,200 | | |
| 32. Steam and exhaust piping | 50,000 | 17,100 | 12,600 | 4,500 | | |
| 33. Distilling plant | 6,800 | 3,600 | 2,500 | 1,100 | | |
| 34. Mechanical telegraphs, counters, gauges, and other instruments | | 2,200 | 4,700 | 3,700 | 1,000 | |
| 35. Heating system | | 2,200 | 1,340 | 1,000 | 340 | |
| 36. Flooring, ladders, gratings, and handrails | | 8,300 | 2,700 | 700 | 2,000 | |
| 37. Safety and stop valve operating gear | | 400 | 200 | 50 | 200 | |
| 38. Spare parts, wrenches, and miscellaneous outfit | | 21,000 | 16,700 | 11,800 | 5,400 | |
| 39. Installation of material | | 7,200 | 7,800 | 300 | 7,800 | |
| 40. Oil heaters and oil-burning apparatus, except pumps | | 14,000 | 4,900 | 3,500 | 1,400 | |
| 41. Electric plant and interior communication | | 36,000 | 28,810 | 18,000 | 5,810 | |
| 42. Reduction gears, case, and main thrusts | | 54,000 | 25,000 | 19,000 | 6,000 | |
| Totals | | 755,950 | 522,000 | 361,550 | 280,450 | |

Total allotment for machinery ..... $522,000
Earned on material (sum of column 5) .....
Earned on labor (sum of column 6) .....
Total earned (labor and material) .....
Amount paid contractors on previous installments on account of machinery .....
Estimated amount due contractors to date on machinery .....
Percentage of completion of machinery .....

.....
*Inspector of Machinery.*

CHIEF OF BUREAU OF STEAM ENGINEERING,
NAVY DEPARTMENT, WASHINGTON, D.C.

Item No. 1: Allowed as a percentage of Items 2 to 41, inclusive.
Item No. 2: Allowed at rate of $3,500 per month for the first 12 months and $800 per month for next 12 months.
Item No. 3: Allowed as follows: 25 per cent upon completion of satisfactory dock trial; 50 per cent upon completion of builders' trials; 25 per cent upon completion of contract trials.
Turbine royalties: 50 per cent to apply on first payment, 25 per cent on eighteenth payment, and balance after official trial.

As the work progresses, the contractors are entitled to request payments through the superintending constructor and the inspector of machinery, and their requests will be approved if the percentage of advancement is such as to warrant payment. The contractor is entitled to credit for material which he has purchased and assigned to a particular vessel and which the inspector has accepted, and for labor performed. In making the request, the contractors submit a statement of their expenditure under these headings. The Bureau's forms showing percentage of completion, the weight report, affidavits in regard to liens and compliance with the provisions of the 8-hour law, and amount earned are submitted by the inspector's office, and the superintending constructor and inspector of machinery make joint recommendation. When the vessel is turned over to the Government at a navy yard a payment is made. The balance of the contract price is paid after the final acceptance, which takes place six (6) months after the date of delivery. Under ordinary conditions the requests occur about twice a month.

Insurance of the vessel and all her appurtenances is provided for in the contract. Policies are forwarded by the contractors direct to the Department, to cover the amount of each payment when it is applied for.

The only insurance the inspector is required to take cognizance of is that relating to sub-contracts. The boilers for our ships are now generally built under a sub-contract. As the contractors request payment on the boilers, each request will be accompanied by a policy of insurance.

## INSTALLATION OF MACHINERY.

The order of installing the principal and auxiliary parts of the different types of engines and boilers with which naval ships are now fitted is substantially the same at the different shipyards and is optional with the contractors. An inspector is, however, usually specially detailed to follow the progress of this work and to note that it is in conformity in workman-

ship and design with the Bureau's specifications and approved drawings.

It is the present practice to install the boilers and their general fittings while the ship is on the ways, and, necessarily, all sea chests and valves and stern sections of propeller shafting will be installed at this time. In the destroyer class of ships all principal parts of the machinery are frequently installed before launching, as is convenient to do so, and as the parts, being comparatively light, may easily be handled with the ordinary appliances of the shipyard adjacent to the building ways.

The heavier parts of battleship engines, especially when of the turbine type, and which require exceptionally powerful cranes to handle and place them, can be more conveniently placed in the ship when she is in the water and alongside a pier. There is also a preference with many engineers for installing the main engines when the ship is water borne, it being considered that better alignment of parts can be secured under such conditions. It is also relevant to mention that the process of installing machinery is a more or less long one, usually extending over about two-thirds of the contract time for the building of a ship.

The main engines and such auxiliaries as air and circulating pumps, prior to being placed in the ship are set up in the shop and the various working connections carefully fitted and adjusted at that time. They are then given brief preliminary trials, driven by steam or compressed air, to prove that they will work freely and smoothly, and, when found to work satisfactorily, are disconnected to such an extent as to be easily handled and sent to the ship for installation in such order of their parts as may be required.

All parts placed on board the ship for installation must have been duly inspected and found to conform with the specifications and drawings and passed, and will, in general, be found to go together or fit into place without particular difficulty; the common practice of setting up and testing the machinery in the shop before sending it to the ship eliminates

many difficulties of fit and adjustment which it would not only be inconvenient but costly to correct while the installation on board ship was in progress.

As a preliminary to installing, if the main engines are of the reciprocating type, the cylinder clearances will be determined by filling the spaces of each cylinder not swept through by the piston, the steam ports, etc., with oil or water and measuring the quantity in volume of the liquid used. If of the turbine type, much care will be taken to secure, as nearly as practicable, a perfect dynamic balance of the rotors. This is accomplished by affixing balancing weights at the lightest points of the rotor on the inside of the drum, these points being located by running the rotor at varying speeds on its shaft, usually by a belt drive from an electric motor. The bearings in which the rotor rests are free to vibrate slightly in response to the varying lateral effort of the rotor, the extent of the vibration and its location for any rate of speed being marked on the rotor shaft by a fixed pointer. Balancing a rotor frequently occupies several days. Following the balancing, the rotor is placed in its casing or cylinder and, if it be of the Parsons type, the dummy and blade peripheral clearances are measured; the former by a permanent micrometer device, the latter by leads.

The balance of the rotor will be further tested by driving it with steam, so as to be fully assured that it will work with the minimum of vibration after it has been installed in the ship. During this final test of its balance the thrust-bearing rings and collars are accurately fitted, and the rotor-shaft bearings scraped to an exact fit to the journal. A test is also made of the peripheral clearance of the blades. Should this not be satisfactory, it may be adjusted by filing off the tips of the blades or by adjustment of the bearings. When in satisfactory condition, a maximum speed test of the rotor is made and its operation carefully observed by an inspector. The rotor is next removed from its cylinder, and its blades and those of the cylinder given a careful final examination for any defects which may have developed during these final tests.

If the turbine is of the Zoelly type, the adjustment of bearings, blade clearances, test of dynamic balance, and final inspection is made substantially as has been described. These operations, however, are generally more easily accomplished in this type of turbine than in the Parsons. The substitution of carbon packing rings as a steam baffle in lieu of the dummy of the Parsons turbine necessitates very careful fitting and adjustment in their installation and much subsequent care to keep them in good working condition.

## BOILER TESTS.

When the installation of the boilers and their immediate attachments is completed, a hydrostatic test is made of each boiler singly to determine its structural strength and tightness, the pressure usually employed being one and one-half times the working steam pressure, and will be found in the machinery specifications. Such defects as become manifest under this test, consisting usually of slight leaks in drum seams, rivet heads, seams and riveting of pads for the usual boiler fittings, and leaks at ends of tubes, are readily corrected. When these defects have been corrected the boilers are subjected to a steam-pressure test, usually about 50 pounds in excess of the working pressure, or as may otherwise be prescribed by the specifications. The test is usually conducted singly or by twos of the boilers.

Should no other defects than slight leaks such as here mentioned manifest themselves, it is not the practice, nor is it desirable, to repeat this high-pressure test. But in cases where the defects are of such a character as to necessitate renewal of a functional part of the boiler, when the renewal is completed the boiler will again be subjected to the maximum steam-pressure test required by the specifications.

## TRIALS.

It is the usual practice, when the machinery installation is sufficiently complete, to make a trial of the machinery with the vessel secured to the dock. The purpose of this trial is to

observe the operation of the machinery under working conditions, to detect any faulty adjustments, as of bearings, to locate air leaks which tend to impair vacuum and to make sure that everything is in good working condition for the preliminary acceptance trials.

The power that may be developed is limited by the strength of the moorings. It is the usual practice to run the main engines at a very moderate speed. In the case of twin-screw ships, one engine will be run in ahead motion, while the other is backing. The engines must be so handled as not to bring a sudden, violent strain on the lines.

Sea conditions are approached as closely as possible. All auxiliaries necessary at sea are operated, thus giving opportunity to observe their adjustment.

When the defects, if any be found, are satisfactorily remedied, the vessel may be reported ready for official trial.

On all builders' trials of a vessel, including dock trials, the inspector of machinery or a commissioned assistant must be present. He will report to the Bureau the results of such trials, with copies of the data obtained if of sufficient importance.

When the contractors for a vessel consider that they are ready for the official speed trial called for under the contract, they will request such trial in writing to the Secretary of the Navy through the superintending constructor and inspector of machinery. Before approving such request it must be shown that the vessel is complete in all respects except final painting and such other work as will be injured on the trial. The inspector of machinery will report to the Bureau regarding such completion, together with his recommendations.

Spare propeller blades will not be cast until after the trial, when the Bureau will be consulted regarding the dimensions of blades desired.

When the board for the trial of a vessel is ordered, the Bureau issues instructions to the inspector of machinery regarding information to be furnished the Trial Board, as follows:

(*a*) A summary of the weights of machinery detailed in groups as per weight instructions; on board and not on board.

(*b*) A certificate stating that the machinery is strong and well built and conforms to contract, plans and specifications and authorized changes.

(*c*) A statement of all items that are not complete.

(*d*) A report of the powers of auxiliaries in use during the trial.

(*e*) A report of test of all gages.

(*f*) A report of setting of safety valves.

(*g*) A copy of synopsis of machinery and hull data.

(*h*) A copy of calibration data, of measuring tanks, torsion meters, etc.

(*i*) A description of any special article or arrangement fitted on a vessel that may be in use during the trial.

(*j*) Such other information of a special nature as may be desired.

The inspector of machinery or an assistant will be directed to report to the trial board and will be present during the trial. He will furnish the board with such information as it may require, in addition to the above named reports, and will afford the board every facility for the examination of the drawings and papers concerning the vessel. During the trial this officer will regard himself as the direct representative of the Bureau. While refraining from any action contrary to the wishes of the board, he will report any circumstances or occurences that he deems of importance direct to the Bureau.

After the trial the inspector may be directed to make the post-trial examination and report results and recommendations to the trial board and Bureau. When post-trial inspection is made by members of engineering trial board the inspector will be present and will afford such officers all assistance and information available.

The nature of the usual post-trial examination will be seen from the following letter, which is taken as an example of the usual practice.

"No. 661. CFH-K

"NAVY DEPARTMENT,

"BOARD OF INSPECTION AND SURVEY FOR SHIPS,

"WASHINGTON.

"*May* 8, 1911.

"From: Senior Member,

"To: Inspector of Machinery, U. S. N.

"Subject: ———; Post-trial examination of machinery.

"1. In accordance with the precept to the Trial Board for the U. S. S. ———, you will direct the contractors to open up and disassemble the following parts of that vessel for inspection:

1. All boilers.
2. Lift all turbine casings sufficiently for examination of blading.
3. One main feed pump.
4. One main air pump.
5. One fire and bilge pump in fireroom.
6. One auxiliary feed pump.
7. One oil pump, each kind.
8. One main circulating-pump engine.
9. Both thrust bearings.
10. Open one blower engine.
11. Fill and test all condensers.

"2. Upon the completion of the examination of these parts you will report to the President of the Board their condition. Should you find any unusual condition of any part of the machinery, you will report immediately upon that part by wire, keeping such part disassembled or open until further orders.

"3. In order not to delay the report going into the Department, examination and report should be made as quickly as practicable.

"(Signed) ——— ———."

## SHOP MANAGEMENT.

There is perhaps no form of shop management that can be more depended upon to secure satisfactory results than close and intelligent supervision.

Day work has been described as the most costly of all workshop systems, and few who have had experience in shop management will attempt to refute the charge; but day work is indispensable, inasmuch as it is the only system under which a large part of the work coming to a marine shop can be satisfactorily done. Machinery repairs can best be managed by day work, even though there be records on file of previous work bearing more or less resemblance to that in hand. The reason of this is that each repair job presents its own difficulties, requiring special expedients and time to overcome. Perhaps the best that can be done in estimating the cost of such work is to consult such records of work of similar kind and extent as may be available and add a percentage for contingencies. This is one of the chief causes for repair work usually overrunning the estimated cost.

Day work is also the system best adapted to the execution of unusual new work for which there are no available records.

There can be no valid excuse urged for failure to apply a bonus, or piece-work system, to the general line of new work with which foreman and workmen are alike familiar, and of which there are abundant reliable records for guidance. It is important to remember, however, that inspections will require to be more frequent and rigid under these systems than under day work; the tendency of the workman being to make the most of his opportunity to secure high wages. The saving under either of these systems, however, when well established will be found to be so marked as to fully justify it and to make the addition in cost for supervision and inspection almost negligible.

The following time and index cards in use in some shops will give a fairly clear conception of the manner of conducting a bonus or premium system.

BONUS SYSTEM.

Form A is a time card given to the workman on his being assigned to the power tool selected for the work, and on it will be found the following data: Number and name of shop or-

der, number of the machine, shop reference symbol, number and name of workman, time of issuing order, drawing number, piece number and number of pieces, operation and finish,

Form A

MACHINE SHOP.

Order No. 297-8 Part Thrust Shaft
Issued 7, 12, 11 No. 74 Name John Doe Symbol J. J. A.

| Drawing Number | Piece Number | No. of Pieces | Operation | Hours of labor. 7/12 10½ | 8 10½ | 9 10½ | 10 10½ | 11 10½ | 12 5 | 13 10½ | 20 10½ | 21 7½ | Total Time | Straight Time | Machine Number |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 14023 | 2 | 2 | Turning and polishing complete. | | | | | | | | | | 84 | 84 | 134 |

Price
Estimated time 112 hours
Time started 7 A.M.
Time finished A.M.

This card must be returned to machine shop office at night.

H. John Jones Inspector.
H. John Smith Foreman.

time and date of starting work and dates during which it was in progress, hours worked daily and total time required to complete work. When completed, the work will be inspected, as indicated on the card, by the shop inspector, the card signed and approved by him and passed on to the shop foreman for approval and signature, when it is returned to the shop office to be audited and filed by the time clerk.

Form B is the filing card which, in addition to being a transcript of the time card, sets forth the premium or bonus earned by the workman, based upon his hourly rating, which is here given as 28½ cents.

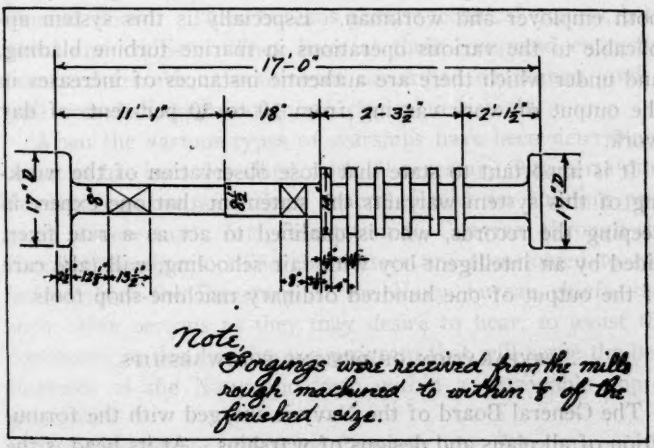
The reverse side of the filing card Form B is a sketch of the object to be machined, showing the general dimensions, and a note as to the condition in which the work goes to the machine, whether a rough forging, rough machined, or whatever the case may be. This shows approximately the amount of metal to be removed to bring the work to the required finish. The object of the data given is to furnish a guide in estimating future work of a similar nature.

Owing to the power tool, and all other accessories neces-

Order No. 297–8
Drawing No. 844023 Operation- turning & threading
Name of Piece- Thrust shaft Material- Wrought Steel
Part of Line shafting

| NO. | | Rate | Date | Tool No. | No. of Pieces | Mark | Est. Time | Act. Time | 50% PROFIT | | Est. Time Each | Act. Time Each | COST EACH | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 74 | John Doe | 28½ | 7 31 | 134 | 2 | | 112 | 84 | $ | 3.99 | .56 | .42 | $ | 131.76 |

FORM B.




FORM B, REVERSED.

sary in doing the work, belonging to the employer, the workman receives but 50 per cent. of his earnings due to time saved as shown.

The total cost of machining the two thrust shafts under the estimated time would have been $31.92, the workman re-

ceiving pay at the rate of 28½ cents an hour; under the bonus system the total cost of machining the shaft is $27.92, a saving of $4.00 to the employer; the bonus paid the workman bringing his hourly pay up to 33¼ cents an hour.

Such a system is practicable in all machine shops where accurate records have been preserved of the various machining operations in the past, and in all cases of shop management it should be made the rule to preserve in systematic form all records of shop work, if not for the purpose of paying a bonus to workmen, at least as a guide in making estimates of the cost of machining similar work.

The bonus system may be also used as a credit system, in which it is found to operate well as a basis for the selection of workmen for advancement.

For machine-shop hand operations a piece-work system may often be adopted and found to work advantageously for both employer and workman. Especially is this system applicable to the various operations in marine turbine blading, and under which there are authentic instances of increases in the output of work varying from 10 to 50 per cent. of day work.

It is important to state that close observation of the working of this system warrants the statement that one expert in keeping the records, who is qualified to act as a rate fixer, aided by an intelligent boy with fair schooling, will take care of the output of one hundred ordinary machine-shop tools.

## PREPARATION OF DESIGNS FOR WARSHIPS.

The General Board of the Navy is charged with the formulation of all plans and designs of warships. At its head is the Admiral of the Navy, its personnel otherwise is made up of the Aides to the Secretary, the Director of Naval Intelligence, President of the Naval War College, and a number of other officers of different grades, noted as students in naval warfare and modern warship design and equipment.

The Office of Naval Intelligence and the Naval War College

are sources from which much important information is obtained, the former supplying advance and confidential information concerning the trend of naval affairs in foreign countries, the latter informing the Board in relation to the practical and military problems worked out at the War College, and the needs of the fleet from the standpoint of strategy and tactics. The Aide for Operations is in possession of all information transmitted from the fleet, as regards tactical and military questions.

The Aide for Material is familiar with all questions relating to the efficiency of the motive appliances, and can co-ordinate information and data supplied by the *material* bureaus.

The Bureaus of Ordnance, Construction, and Engineering supply the General Board with information along definite lines coming under their cognizance, and likewise with the preparation of plans that will conform to the desire of the General Board. These bureaus possess technical experts, both naval and civilian. They have at their command extensive tabulated records, a contingent of skilled draftsmen, and a well rounded organization of clerical assistants.

When the various types of warships have been determined upon, as has been described, with all necessary data pertaining thereto, together with their general plans, the building program is now ready for presentation to Congress through the Naval Committees. The committees are accustomed to call before them the Secretary of the Navy, bureau chiefs, and such other persons as they may desire to hear, to assist the committee in deciding upon a program that will serve the best interests of the Navy and keep within a reasonable appropriation.

Upon favorable action by Congress, the Secretary of the Navy may proceed to advertise for bids for the construction of the authorized ships.

A " Circular of requirements for bidders" is issued to shipbuilders, together with the Department's specifications and type plans. Bidders may submit proposals based upon exact compliance with the Department's plans and specifications, or

upon their own plans for motive power, conforming in general to the Department's plans. A date is set for the opening of all proposals, and the contract is awarded to the lowest responsible bidder.

Immediately after the contract for the building of any warship is made, tentative plans are prepared for further construction. The plans are practically elaborated by the Bureaus of Construction and Steam Engineering as the work progresses. The work of these bureaus overlap in so many directions that numerous conferences and compromises have to be effected before a final plan is agreed upon.

The final plan of a modern battleship represents so many compromises that it may be truthfully said that the particular desires of the General Board are seldom secured, except as regards as to the speed and the size of the main battery, in which respect its demands are seldom opposed.

# MARINE FEED-WATER HEATING.

By Lieutenant Commander H. C. Dinger, U. S. N.

## METHODS OF REDUCING STEAM CONSUMPTION IN MARINE ENGINEERING.

The saving of fuel is an important matter to all those interested in Marine and Naval Engineering. The methods of securing economy aside from avoiding pure waste entailed by leaks, loss of fuel and bad firing revolve themselves about the question of the handling of heat. A pound of fuel contains so many heat units which may by various means be converted into useful mechanical energy. The methods of handling and applying this heat may cause an increase or a decrease in economy, but the methods of handling the heat are the basic processes in which improvement in economy may be sought.

No matter whether it is steam with a turbine or a reciprocating engine, a gas or an oil engine, the electric drive or a reduction gear, the basis of success in securing improvement in economy is in the method and process of handling the heat. Anything that causes heat units to be wasted or lost without being converted into work will reduce economy and, on the other hand, anything that saves a heat unit from being wasted accomplishes a gain in economy. Economy of fuel is, after all, a thermodynamic problem, and success is only to be gained by paying due regard to and following out proper thermodynamic principles.

Marine engineers are satisfied with a vacuum that the up-to-date power plant would consider ruinous. Superheating is

almost unknown. Feed heaters are frequently not employed or are not properly operated for full efficiency. Yet the item of fuel is the big item in the operation of steamships. Why these important aids to economy are so almost utterly neglected it is difficult to understand, except that first cost is considered too much. By proper attention to securing the utmost possibilities in these fields the consumption of fuel can be reduced from the current 1.5 pounds per I.H.P. to 1.2 pounds per I.H.P. without change in the engine. Different designs of engines vary the economy, but the possibilities to be gained by the use of superheat, high vacuum and highest feed temperatures are greater than changes in engine design are likely to produce for several years to come.

## FEED HEATING.

Feed heating is a most important matter in the campaign of economy of fuel. Feed heaters are now generally used on practically all marine steam engined vessels, but it is extremely doubtful whether the great effect that their successful operation has upon economy is fully realized by many engineers. It is also known that the full gain from the use of properly designed feed-water heaters is not being secured on many marine craft and that present marine engineering practice has failed to fully utilize the full possibilities of feed-water heating.

The following notes give a resumé of the general principles underlying the matter of feed heating for naval and marine work and deal largely with the practical points of the construction and operation of feed heaters. The information has been gathered from various available sources, and most of the practical points mentioned are the result of observation based on actual experience.

The discussion of the principles and points of operation is made on a rather elementary basis, with the idea of making these notes of practical service to those actually having con-

trol of the operation and installation of this apparatus. The possibilities of fuel saving will not be fully realized unless the personnel actually controlling the operation understand how the saving is accomplished and know what the conditions are that militate against it.

The purpose of the boiler feed-water heater in marine service is twofold: (1) as a means for securing less strain and distortion on the boilers and (2) as a means for securing economy of fuel consumption and saving heat units that otherwise would be wasted.

In marine engineering practice it is usual to carry a fairly good vacuum, and the temperature in the hotwell or air-pump discharge is therefore ordinarily about 100 degrees F. If there is no feed heating, this water is discharged to the boiler, where, if the working pressure is 200 pounds, the temperature will be 387 degrees F.

The entrance of feed water at 100 degrees into the boiler shell or drum where the water is 287 degrees will cause a chilling effect and considerable disturbance near the points where the feed is released, and will create wide differences in temperature of the water in different parts of the boiler. This has the effect of causing distortion and the setting up of strains in the material of the boiler, which are liable to cause leakage. The wide difference of temperature of the entering feed from that of the water in the boiler also disturbs the free steaming qualities of the boiler and checks evaporation. Just how this is done is not exactly known, but it is probable that counter currents and other disturbances are set up which detract from the free formation of steam.

The ideal condition, of course, would be to have the feed water at the temperature of the water in the boiler. There would then be the least possible disturbance and no strain whatever produced by differences in the temperature of the water.

It is therefore very desirable to have warm feed water, on account of lessening strain on the boiler and because it improves the steam-forming qualities of the boiler.

The importance of heating feed water before delivery to the boiler on the ground of economy can be best realized by considering what takes place in the exchange of heat in the generation of steam. Steam at any pressure has a *temperature* corresponding to that pressure, and also a certain *Total heat, Latent heat* and *Sensible heat* corresponding to that pressure. In the generation of steam the water is first heated up to the temperature corresponding to the pressure at which it is evaporated. The amount of heat that the water contains on the point of evaporation at any pressure is the sensible heat. This corresponds almost exactly to the temperature above 32 degrees F. That is, water of 387 degrees temperature has approximately (387—32) equals 355 heat units per pound. When the water has been heated to the temperature of evaporation and more heat is added, it will begin to evaporate. The amount of heat required to convert water into steam at any temperature is the *latent heat.* The *latent heat* plus the *sensible heat* is the *Total* heat of the steam. The *Latent heat* varies with the temperature, being less as the temperature and pressure increase.

If the feed water enters a boiler at 100 degrees F., to be evaporated at 200 pounds pressure, it will first have to be raised to 387 degrees F., thus requiring (387—100) equals 287 heat units per pound and, then 839 heat units for latent heat; in all 1,126 heat units.

If, however, the feed water had been heated to 200 degrees, only (387—200 + 839) or 1,026 heat units would have been required, the heat saved in this case being about nine per cent. The percentage of saving by feed heating, considering that the steam used as heating agent is waste steam, will increase as the temperature of feed is increased; being roughly one per cent. for every ten or eleven degrees of feed temperature. The per cent. of saving will also decrease as higher boiler pressures are used.

The percentage of heat saved by heating feed water with boiler pressure at 100 pounds is shown in the following table:

PERCENTAGE SAVING IN FUEL BY HEATING FEED WATER WITH EXHAUST STEAM.—100 POUNDS BOILER PRESSURE.

| Initial temperature of feed water. | Final temperature of feed water. | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 120 degrees. | 130 degrees. | 140 degrees. | 150 degrees. | 160 degrees. | 170 degrees. | 180 degrees. | 190 degrees. | 200 degrees. | 210 degrees. | 220 degrees. | 230 degrees. |
| °F. | P. ct. | P. ct. | P. ct. | P. ct. | P. ct. | P. ct. | P. ct. | P. ct. | P. ct. | P. ct. | P. ct. | P. ct. |
| 40 | 6.8 | 7.65 | 8.72 | 9.35 | 10.45 | 11.05 | 12.20 | 13.07 | 13.95 | 14.80 | 15.70 | 16.55 |
| 50 | 5.95 | 6.85 | 7.92 | 8.57 | 9.70 | 11.28 | 11.45 | 12.32 | 13.20 | 14.10 | 15.00 | 15.85 |
| 60 | 5.1 | 6.04 | 7.12 | 7.57 | 8.90 | 9.50 | 10.70 | 11.60 | 12.45 | 13.35 | 14.25 | 15.15 |
| 70 | 4.25 | 5.23 | 6.27 | 6.97 | 8.07 | 8.72 | 9.87 | 10.75 | 11.65 | 12.55 | 13.45 | 14.35 |
| 80 | 3.4 | 4.39 | 5.43 | 6.15 | 7.24 | 7.91 | 9 05 | 9.95 | 10.85 | 11.75 | 12.70 | 13.60 |
| 90 | 2.55 | 3.54 | 4.57 | 5.32 | 6.40 | 7.09 | 8.22 | 9.13 | 10.05 | 10.95 | 11.90 | 12.80 |
| 100 | 1.7 | 2.68 | 3.68 | 4.47 | 5.53 | 6.26 | 7.38 | 8.30 | 9.22 | 10.15 | 11.05 | 12.00 |
| 110 | 0.85 | 1.8 | 2.78 | 3.61 | 4.65 | 5.41 | 6.52 | 7.43 | 8.38 | 9.30 | 10.25 | 11.15 |
| 120 | ... | 0.91 | 1.88 | 2.73 | 3.75 | 4.55 | 5.63 | 6.58 | 7.50 | 8.45 | 9.40 | 10.30 |
| 130 | ... | ... | 0.95 | 1.84 | 2.84 | 3.68 | 4.74 | 5.68 | 6.63 | 7.60 | 8.55 | 9.45 |
| 140 | ... | ... | ... | 0.92 | 1.90 | 2.98 | 3.83 | 4.80 | 5.75 | 6.70 | 7.65 | 8.60 |
| 150 | ... | ... | ... | ... | 0.97 | 1.97 | 2.90 | 3.85 | 4.83 | 5.80 | 6.75 | 7.72 |

The per cent. of saving per degree of temperature for different conditions is given in following table, taken from Kent's "Engineer's Pocket Book".

An approximate rule for the condition for ordinary practice is a saving of one per cent. is made by each increase of 11 degrees in the temperature of the feed water. This corresponds to .0909 per cent. per degree.

The calculation of saving is made as follows: Boiler pressure 100 pounds, gage; total heat in steam above 32 degrees equals 1,185 B.t.u. Feed water, original temperature 60 degrees, final temperature 209 degrees F. Increase in heat units, 150. Heat units above 32 degrees in feed water of original temperature equals 28. Heat units in steam above

FEED-WATER HEATER.

**Percentage of Saving for each Degree of Increase in Temperature of Feed Water Heated by Waste Steam.**

| Initial temp. of feed. | Pressure of steam in boiler; pounds per square inch above atmosphere. | | | | | | | | | | | Initial temp. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0 | 20 | 40 | 60 | 80 | 100 | 120 | 140 | 160 | 180 | 200 | |
| Degs. | | | | | | | | | | | | |
| 32 | .0872 | .0861 | .0855 | .0851 | .0847 | .0844 | .0841 | .0839 | .0837 | .0835 | .0833 | 32 |
| 40 | .0878 | .0867 | .0861 | .0856 | .0853 | .0850 | .0847 | .0845 | .0843 | .0841 | .0839 | 40 |
| 50 | .0886 | .0875 | .0868 | .0864 | .0860 | .0857 | .0854 | .0852 | .0850 | .0848 | .0846 | 50 |
| 60 | .0894 | .0883 | .0876 | .0872 | .0867 | .0864 | .0862 | .0859 | .0856 | .0855 | .0853 | 60 |
| 70 | .0902 | .0890 | .0884 | .0879 | .0875 | .0872 | .0869 | .0867 | .0864 | .0862 | .0860 | 70 |
| 80 | .0910 | .0898 | .0891 | .0887 | .0883 | .0879 | .0877 | .0874 | .0872 | .0870 | .0868 | 80 |
| 90 | .0919 | .0907 | .0900 | .0895 | .0888 | .0887 | .0884 | .0883 | .0879 | .0877 | .0875 | 90 |
| 100 | .0927 | .0915 | .0908 | .0903 | .0899 | .0895 | .0892 | .0890 | .0887 | .0885 | .0883 | 100 |
| 110 | .0936 | .0923 | .0916 | .0911 | .0907 | .0903 | .0900 | .0898 | .0895 | .0893 | .0891 | 110 |
| 120 | .0945 | .0932 | .0925 | .0919 | .0915 | .0911 | .0908 | .0906 | .0903 | .0901 | .0899 | 120 |
| 130 | .0954 | .0941 | .0934 | .0928 | .0924 | .0920 | .0917 | .0914 | .0912 | .0909 | .0907 | 130 |
| 140 | .0963 | .0950 | .0943 | .0937 | .0932 | .0929 | .0925 | .0923 | .0920 | .0918 | .0916 | 140 |
| 150 | .0973 | .0959 | .0951 | .0946 | .0941 | .0937 | .0934 | .0931 | .0929 | .0926 | .0924 | 150 |
| 160 | .0982 | .0968 | .0961 | .0955 | .0950 | .0946 | .0943 | .0940 | .0937 | .0935 | .0933 | 160 |
| 170 | .0992 | .0978 | .0970 | .0964 | .0959 | .0955 | .0952 | .0949 | .0946 | .0944 | .0941 | 170 |
| 180 | .1002 | .0988 | .0981 | .0973 | .0969 | .0965 | .0961 | .0958 | .0955 | .0953 | .0951 | 180 |
| 190 | .1012 | .0998 | .0989 | .0983 | .0978 | .0974 | .0971 | .0968 | .0964 | .0962 | .0960 | 190 |
| 200 | .1022 | .1008 | .0999 | .0993 | .0988 | .0984 | .0980 | .0977 | .0974 | 0972 | .0969 | 200 |
| 210 | .1033 | .1018 | .1009 | .1003 | .0998 | .0994 | .0990 | .0987 | .0984 | .0981 | .0979 | 210 |
| 220 | ... | .1029 | .1019 | .1013 | .1008 | .1004 | .1000 | .0997 | .0994 | .0991 | .0989 | 220 |
| 230 | ... | .1039 | .1031 | .1024 | .1018 | .1012 | .1010 | .1007 | .1003 | .1001 | .0999 | 230 |
| 240 | ... | .1050 | .1041 | .1034 | .1029 | .1024 | .1020 | .1017 | .1014 | .1011 | .1009 | 240 |
| 250 | ... | .1062 | .1052 | .1045 | .1040 | .1035 | .1031 | .1027 | .1025 | .1022 | .1019 | 250 |

that is cold feed water, 1,185—28 equals 1,157. Saving by the feed-water heater equals 150÷1,157 equals 12.96 per cent. The same result is obtained by the use of the table. Increase in the temperature 150 degrees times tabular figure .0864 equals 12.96 per cent. Let total heat of 1 pound of steam at the boiler pressure equal H; total heat of 1 pound of feed water before entering the heater equal $h_1$, and after passing through the heater equal $h_2$; then the saving made by the heater is

$$\frac{h_2 - h_1}{H - h_1}.$$

There is an increase in economy even if the live steam from the boiler is used as the heating agent, and live-steam feed heaters are sometimes used. There is, however, much more

economy if waste steam or steam that has had some of its heat spent is used as the heating agent.

In marine steam-engine plants the main engines are designed to operate so as to use the steam in the most economical manner from the initial pressure down to the highest vacuum that can be maintained. The steam is used expansively, and if a vacuum of 27 inches is secured the temperature of the condensed steam, on its arrival at the feed tank, will be about 110 degrees F. There are, however, numerous auxiliaries on board ship that do not use the steam expansively and which would use about the same amount of steam whether the exhaust was at atmospheric pressure or at a high vacuum. Such auxiliaries as the reciprocating pumps, circulating engines, hoisting engines, etc., come in this category. There are also numerous drains from steam pipes, valve chests of main and auxiliary engines, evaporator coils, heating systems, etc., that contain a mixture of steam and water that have temperatures varying from 400 degrees to 200 degrees. This heat ordinarily would be lost if drained into bilge or condenser.

Now if the heat contained in the exhaust from the auxiliaries and in the drains, etc., is collected and used as the heating agent in a feed-water heater, it will represent almost a clear gain, because heat that would otherwise be lost is then efficiently used.

The heat from the drains will produce a clear gain; that from the exhaust of the auxiliaries, nearly a clear gain. In other words, the auxiliaries would use slightly less steam if they were allowed to exhaust to the condenser instead of discharging to a feed heater, operated under a pressure slightly above the atmosphere.

The great point in this matter is that the condition which causes the saving by the use of feed heaters is that in the feed heater practically all the *latent heat* of the heating steam is used. When an engine exhausts to a condenser most of the latent heat is still left in the vapor, and this heat is absorbed by the circulating water and pumped overboard. But when

the steam, or a portion of it, is drawn off and used in a feed-water heater, the vapor is condensed and the *latent* heat is used to heat the water. The amount of latent heat thus kept from being lost in the condenser represents a clear saving.

Now if it were possible to use all of the latent heat, both in the main exhaust and in the auxiliary exhaust, there would be an enormous saving, and we might even dispense with the circulating water. Unfortunately, temperature comes into play, and, as heat will flow only toward the lower temperature, we can only use a portion of the *latent* heat for feed heating. The feed naturally cannot be heated any higher than the temperature of the heating agent; hence if the water leaves the condenser at the temperature of the exhaust the heat in the exhaust cannot be available since its temperature is too low. But if some of the steam is drawn off from the system at various stages, when its temperature is above that of the exhaust, and used in a feed heater, its *latent* heat, which is otherwise discharged to the condenser, can be saved and used in heating the feed.

When the auxiliary exhaust is used for feed heating the pressure at which it is operated is usually at about the atmosphere. It is deemed advisable in all cases to use a pressure a little above the atmosphere, so as to have the auxiliary exhaust system under a slight pressure, thus excluding the air, which is a chronic and persistent enemy of the engineer, in connection with condensing and feed-heating apparatus. A pressure very much above the atmospheric cannot be used on account of the danger of causing water hammer in the cylinders of the auxiliaries and an undersirable amount of leakage in the auxiliary exhaust pipe joints. Pressures as high as 40 pounds absolute have, however, been successfully used, and the feed water obtained correspondingly warmer.

The amount of heating steam required to heat the feed to the temperature of the auxiliary exhaust can easily be determined from the steam tables. Suppose we have a feed-tank temperature of 100 degrees F. and a back presure of 20 pounds absolute, or 5 pounds above the atmosphere. The temperature

corresponding to 20 pounds absolute is 227 degrees F. Therefore (227—100) or 127 heat units must be supplied to each pound of feed water. The total heat of steam at 20 pounds absolute, above 227 degrees F. is 1,151 B.t.u. It will therefore require .11 pound of steam, as heating agent, to give a temperature of 227 degrees F. Therefore to secure 220 degrees F. feed temperature about 11 per cent. of total weight of steam must be sent to the feed heater. The ordinary auxiliary exhaust and the accumulation from the drains, etc., is about this amount. Where there are relatively few steam-driven auxiliaries the auxiliary exhaust may be insufficient to secure the full temperature. The vapor from the evaporators, for make-up feed, may be used to increase the supply. On the other hand, if there are more auxiliaries in operation and if these are particularly wasteful, there may be more auxiliary exhaust present than enough to heat the feed water. In this latter case some of the auxiliary exhaust can be diverted into the L.P. receivers of the engine, and made to do work there.

If there is an insufficient supply of heating steam the necessary amount may be made up by opening more drains from the steam lines or cylinders. It is more economical to do this than to have a separate live-steam connection to feed heater; since by means of the drains wet steam and not dry steam is drawn off. Or steam from some of the engine receivers may be drawn off and used in the feed heater.

## GENERAL TYPES OF FEED HEATERS.

Heaters may be open (jet) or closed (surface), according as the water and the steam are allowed to mix or not. The open heater is very simple and is also quite efficient. In the merchant service, where a favorable high location for it can usually be obtained, it is largely used.

The open heater, a cut of which is shown, Fig. 1, acts like a jet condenser, all the auxiliary steam entering, it being condensed by the cascade of feed water. This heater must necessarily be situated on the suction side of the feed pump, and

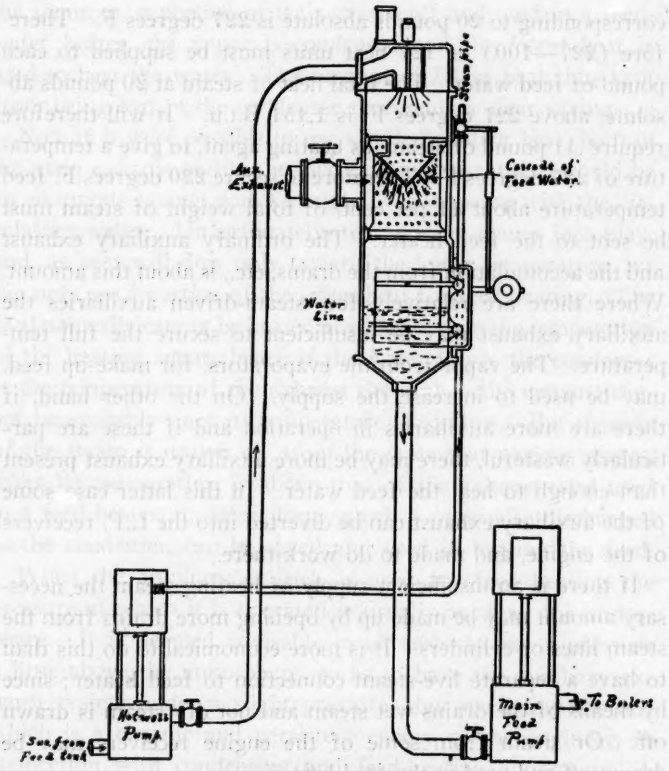



FIG. 1.—OPEN HEATER AND PIPING.

in order to get a sufficient head to prevent the formation of vapor it must be located well above the pump. It is to some extent on account of this necessity for having it up high that this type it not used on naval vessels. To pump the feed water from the hotwell or feed tank a low-duty hotwell pump is provided. The connections for pumping and attachments are shown in cut. The heaters are usually fitted with an automatic float governor for governing the speed of the hotwell pump, so that the water is kept at a nearly constant level in the heater.

With a jet heater the temperature of feed can not be much

above 180 degrees F., as a higher temperature will cause vapor to form, and then the feed pump will not have a proper suction.

For naval vessels heaters are generally of the closed, surface, type; exhaust steam entering the shell outside of the tubes and feed water going through the tubes. Both vertical and horizontal heaters are used. They are installed in two general systems, one where the heater is on the suction, and the other where it is on the discharge side of the feed pumps, Fig. 2 and 3. These are sometimes designated as high-pressure and low-pressure heaters.

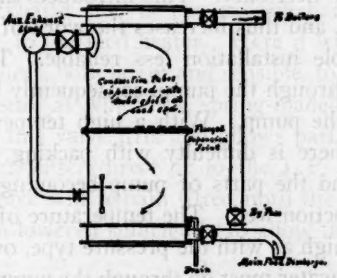


FIG. 2.—CLOSED HEATER ON DISCHARGE SIDE OF FEED PUMP.

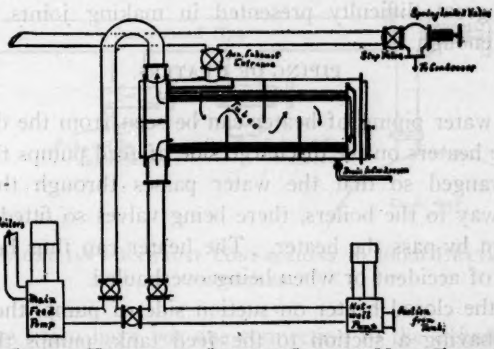


FIG. 3.—CLOSED HEATER ON SUCTION SIDE OF MAIN FEED PUMP.

With heater on the discharge side of the pump the feed, when passing through the pump, is not warm, hence there is no difficulty in suction, due to vapor. No other pump, beside the

feed pump, is needed, and the weight and the cost of the hotwell pump is saved. The temperature can be maintained at a temperature very close to that of the exhaust steam, there being no limits due to heating of pump suction.

The disadvantages are: The parts of the heater have to be made extra heavy to withstand the pressure. Where high pressure is used it is difficult to secure tight joints, and unless very good materials are used and good workmanship present there is danger of gaskets blowing out.

With heater on the suction side of pump the addition of a hotwell pump is necessary. This introduces another auxiliary to be cared for, and thus increases the work of attendance and makes the whole installation less reliable. The feed is hot while passing through the pump, consequently bringing harder conditions on the pump. With a high temperature of water in the pump there is difficulty with packing, vapor forming in the pump and the parts of pump becoming heated so that they do not function well. The temperature of feed can probably not be as high as with the pressure type, owing to the fact that the heated water must go through the pump. On the other hand, everything about the heater being of low pressure, there is not great difficulty presented in making joints or parts strong enough.

## PIPING OF HEATERS.

The water piping of heater can be seen from the diagrams. For the heaters on the discharge side of feed pumps the valves are arranged so that the water passes through the heater on its way to the boilers, there being valves so fitted that the feed can by-pass the heater. The heater can thus be cut out in case of accident or when being overhauled.

For the closed heater on suction side of pump, the hotwell pump, having a suction to the feed tank, pumps the water through the heater (or by suitable arrangement of valves can be by-passed) and to a feed-pump suction. In this way the feed pump has a forced suction, and the liability of having vapor in the pump suction is overcome.

## STEAM PIPING CONNECTIONS.

The steam-piping connection should be arranged so that all the auxiliary exhaust steam comes to the heater and that it is condensed there (dead-end heater). Adjustable relief valves should be fitted at the connection of the auxiliary exhaust pipe to the condenser so that the pressure in the auxiliary exhaust can be regulated. Connections can be made so that the exhaust steam can, when desired, be passed through the heater on its way to the condenser.

In the dead-end heater, as long as the desired pressure in the auxiliary exhaust line is not exceeded, all the exhaust steam will come to the feed heater. Here it will be condensed and give up its heat, both latent and sensible, to the feed water. When the pressure at which the spring-loaded relief valve is set is exceeded this valve lifts and allows part of the exhaust to go to the condenser direct or to the L.P. receiver of the engine. The valve will remain lifted until the pressure in the exhaust is again lowered sufficiently to allow the valve to seat, Fig. 4.

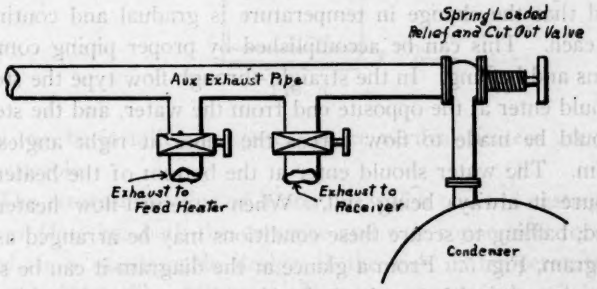



FIG. 4.—AUXILIARY EXHAUST CONNECTIONS TO FEED HEATER AND CONDENSER.

Heaters often have but one connection to the exhaust pipe. If a second connection is fitted nearer to the condenser and a cut-out valve fitted between, part of the auxiliary exhaust steam can be passed through the heater on its way to the condenser. This latter may sometimes be desired in port. This

connection also insures that a portion of the auxiliary exhaust must pass through the heater, even if the valve on condenser is open, thus producing a positive flow of exhaust steam through heater, and preventing heater from becoming air bound on steam side. Heaters are fitted both with and without this sesond connection to auxiliary exhaust line. The connection adds somewhat to complication and is not essential or necessary to the operation of the heater.

The pressure on the heater may also be regulated by having a relief valve on the heater and the discharge from this valve led to the L.P. receiver or condenser. As, however, it is necessary to have the auxiliary exhaust connect with the condenser, the fitting of the relief valve on the auxiliary exhaust connection to condenser serves the purpose and may require less piping. See Fig. 14.

### POSITION OF STEAM AND WATER CONNECTIONS, BAFFLING.

To secure the hottest temperature of feed the heater should be arranged so that the hottest water strikes the hottest steam and that the change in temperature is gradual and continual in each. This can be accomplished by proper piping connections and baffling. In the straight-through-flow type the steam should enter at the opposite end from the water, and the steam should be made to flow across the tubes at right angles to them. The water should enter at the bottom of the heater to ensure it always being full. When a return-flow heater is used, baffling to secure these conditions may be arranged as in diagram, Fig. 5. From a glance at the diagram it can be seen that the vertical heater lends itself with greater facility to the above requirements, Fig. 6. Other advantages of this type are (1) less weight than the return-flow type, owing to smaller end castings; (2) simpler baffling. The horizontal return-type does not lend itself as well to proper requirements. In all horizontal types there is the liability of upper rows of tubes becoming filled with air. The horizontal types do, however, in many cases provide greater facility for the removal of tubes

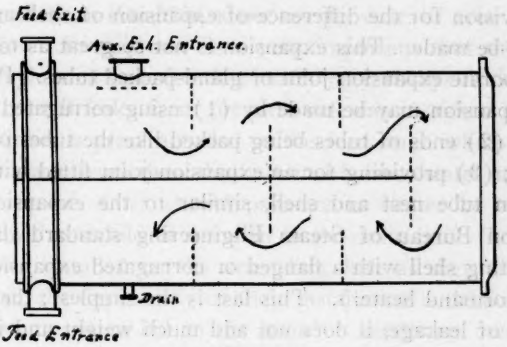



FIG. 5.

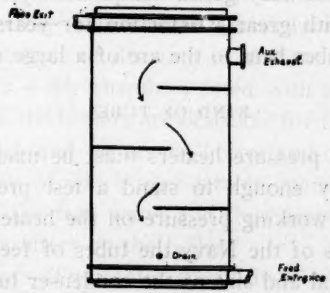



FIG. 6.

and can be installed in spaces where vertical heaters are impracticable.

The return-flow type of heater, in which one end casting or cover plate has both the entrance and exit connection for the feed, is a questionable design if the rise in temperature is considerable. Trouble is usually experienced in keeping the joints tight and also in cracking of tube sheets. When so arranged a portion of the tube sheet or end casting is subjected to the temperature of the entering water, while the other half or portion is subject to the temperature of the leaving feed. This difference of temperature may be as much as 150 degrees, and it can readily be seen that distortion of the parts and leakage and breakage may be expected under these conditions.

Provision for the difference of expansion of shell and tubes should be made. This expansion is not so great as to require an elaborate expansion joint or gland-packed tubes. Provision for expansion may be made by (1) using corrugated or bent tubes; (2) ends of tubes being packed like the tubes of a condenser; (3) providing for an expansion joint fitted with gland between tube nest and shell, similar to the expansion joint fitted on Bureau of Steam Engineering standard distillers; (4) fitting shell with a flanged or corrugated expansion joint (see Normand heater). This last is the simplest; there is no danger of leakage, it does not add much weight and is easily made. The difference of expansion between shell and tubes is not likely to be very great. Expansion joints of this type have worked with great satisfaction for years. Another device is to use tubes bent to the arc of a large circle.

### KIND OF TUBES.

The tubes of pressure heaters must be made of good material and heavy enough to stand a test pressure of about three times the working pressure on the heater. On most of the large vessels of the Navy the tubes of feed heater are of the same material and size as the condenser tubes. For pressure heaters composition tubes 5/8 inch diameter, 16 B.W.G., are largely used. For suction heaters a lighter gage tube is used. Steel tubes are also used, and, if fresh water is assured, their lasting qualities should be satisfactory. Steel tubes are largely used on the Great Lakes. Copper tubes are supposed to be more efficient than brass or bronze tubes, and where the tubes are corrugated or made into spirals this material is largely used. Copper tubes are usually tinned to guard against corrosion due to hot water.

### FITTINGS ON HEATERS.

*Drain.*—The size of drain should be proportioned to the percentage of total steam used in the feed heater. In heaters heating to about 200 degrees F. by use of the auxiliary exhaust,

the area of drain should be about ten per cent. of area of feed connection. In any case it is very important that the drain be of ample area in order that heater may always drain easily and that small leaks from pressure side can be taken care of.

The drain is preferably led to the feed tank, which should be at a lower level than the bottom of feed heater to avoid backing up of water should pressure in heater be reduced. A check valve should be installed in the drain line. Traps are sometimes installed on the drain line, but, as they are often unreliable, the desirability of their installation is debatable.

Drains are often led to the main or auxiliary condenser. When led to condenser the heat in the drain water is lost, while if drain is led to the feed tank its heat may be saved. It is also objectionable to lead these drains to main condensers on account of danger of drawing in air.

*Air Chambers.*—Air chambers fitted with pet cocks on the discharge end of the heaters are desirable for the double object of reducing shock and jar on the heater and of serving as an air trap by means of which considerable quantities of air may be taken out of the feed water and thus prevented from having its corrosive effect on the boiler.

*Other Fittings.*—The heater should have a compound pressure gage connected to steam side, a glass water gage with try-cocks. An air cock arranged to drain air from the space immediately above the water level should be supplied. This last is one of the most essential fittings for securing efficient operation of feed heater.

## HEATING SURFACE REQUIRED.

The amount of heating surface to allow depends largely upon the design of heater. The following, however, is an approximation of the amount that may be permitted to secure 100 degrees rise in temperature.

For plain steel or composition tubes without retarders and not carefully baffled on steam side allow one square foot of steam surface per 280 pounds of feed per hour.

For well-baffled plain composition or copper tubes fitted with efficient whirling plates or retarders, and with steam side carefully baffled, allow one square foot of steam surface for 350 pounds of water. Type, Normand or Wainwright plain-tube heater.

For spiral coils of copper or spiral corrugated tubes with proper steam baffling allow one square foot of steam surface per 400 pounds of water. Types, Reilly heater, Wainwright corrugated-tube heater.

For spiral corrugated-film heaters with steam well baffled allow one square foot of steam surface for 750 to 800 pounds of water. Type, Shutte & Koerting film heater.

For rise in temperature greater or less than 100 degrees the surface may be proportioned approximately according to the temperature rise.

*Size of Tubes.*—The size of tubes used for feed heaters affects the weight, cost and efficiency of the heater.

The smaller the tube the less weight and greater the efficiency of transmission. The smaller-size tube will, however, greatly increase the friction; hence tubes should be sufficiently large so as to avoid any undue friction. With small tubes the thickness of metal may be much less for the same strength. Thickening the tube wall will decrease the rate of heat transmission.

It is believed that tubes as small as ½ inch may be used without unusual friction, though the sizes most generally used are ⅝ inch and ¾ inch. If the size of tubes is decreased too much there is a corresponding increase in the work of fitting them and securing tight joints. If tubes are corrugated the friction will be increased, and there is also danger of the metal becoming brittle in the process of corrugation and later cracking in service.

THE RELATIVE DIRECTION OF FLOW OF THE STEAM AND WATER.

Heaters should be arranged so that the feed leaves near the entrance point of the heating steam. The limit of the temperature attainable is fixed by the temperature of the heating steam;

hence, in order to secure the highest temperature, the feed must leave at the point where there is the hottest steam. It is therefore plain to see that the water and the steam should flow in opposite directions, counter-current principle. This is very simple, yet in many heaters this principle is violated. In some designs the steam is allowed to enter almost anywhere.

In plain tube heaters the flow of the steam across the tubes is usually guided by baffles. The area for the passage of steam should be gradually decreasing. Better heating effect is secured if tubes are staggered and if the flow of steam is approximately normal to the tubes carrying the water. Too much baffling is undesirable, and if proper conditions in regard to flow of steam and water can be secured without baffling so much the better.

The steam connection should have a perforated baffle or scattering plate fitted in its wake to prevent the direct impingement of the blast of steam upon the tube.

### THE FLOW OF WATER THROUGH THE HEATER.

The length of time or the distance that the water should be in contact with the heating steam is a matter of debate. For conditions met with in marine and naval work the distance should not be less than ten feet or more than forty feet for every 100 degrees F. rise in temperature. The efficiency of heat transmission increases as the relative velocity of heated and heating mediums is increased.

There are various devices and arrangements that affect the efficiency of the transmission.

A plain, straight tube is not particularly efficient. A bent tube or a tube with a twisted baffle retarder is more efficient. A spiral coil or a spirally-corrugated tube are still more efficient. In a straight tube the water flows through and the column of water is heated only on its surface. A bent tube causes some breaking up of this column and causes more particles of water to come into contact with the heating surface. A spiral coil or a spiral-corrugated tube or a spiral retarder gives the water

a whirling movement and sets up centrifugal motion, which acts to cause the cooler water at the center of the column to move out to the surface, thus setting up a circulation in each tube and causing all the particles to come into contact with the heating surface. This whirling movement is very essential to secure the best results. It is obtained to a high degree in the corrugated-film type of heater.

If possible, the tube element should be arranged so that it is vertical. This eliminates the effect of gravity in modifying the whirling motion, it is less liable to allow air bubbles to collect on the heating surface, gives a better drainage to the condensed vapor on the steam side and guards against the accumulation of scale and dirt.

Spiral baffles or spiral tubes will add about 30 per cent. to the efficiency of the heating surface over the plain tube. Employing corrugated films will more than double the efficiency of the surface.

The entrance head or chamber should be fitted with a perforated baffle or scattering plate to cause the feed water to go to all the tubes and not merely flow through the nearer ones.

## OPERATION OF HEATERS.

The operation of a feed heater is a very simple thing and is generally so regarded, yet in practice many feed heaters do not give the temperatures of feed that they are capable of giving when properly handled.

If there is sufficient heating steam, and the capacity of heater is not overtaxed, the heater should give a temperature within about five degrees of the temperature of the heating agent.

The principal reason why feed heaters do not give a proper temperature is the presence of air on the steam side. The exhaust steam contains a certain amount of air. This air, being heavier than steam, will collect above the water level in the heater and if it is not drawn off it will gradually displace the steam and will render a large portion of the heating surface inactive.

All heaters should be provided with means of drawing off this air just above the water level. The air connection must be at this point, so that it will draw off the air and not the steam. The air cock should be left open as long as air issues; when water or steam comes out it should be closed off.

The way the air collects above the surface of the water is illustrated in Fig. 6a. It will be noted that the air vent must

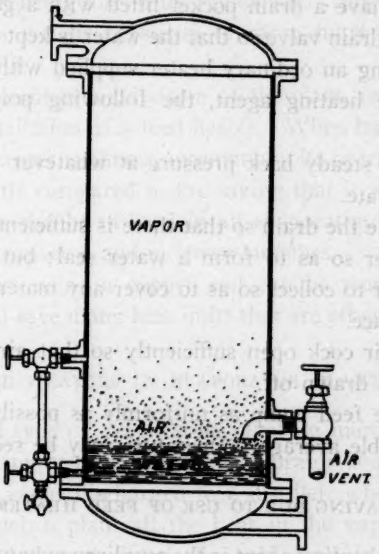



FIG. 6A.

be placed at a point above the level of the water, and yet not so high above the water that vapor instead of air is drawn off.

This matter of air collecting in feed heaters is a simple and natural matter, yet a large percentage of our operating engineers do not appear to know anything about it, and a very large number of feed heaters are built without having a proper provision made for drawing off this air.

Another cause for improper operation is allowing the heater to fill up with water. In this case the water displaces the steam and renders the heating surface inactive. The remedy

for this is to regulate the drain valve so that proper drainage is secured. In doing this, however, there is another trouble encountered, that is the drain is opened too much, thus allowing the steam to blow through into the drain and to feed tank or condenser. This, of course, means a loss of the heating steam. To properly regulate the drain a steam trap may be employed, but a trap cannot always be trusted, and usually it is better to have a drain pocket fitted with a gage glass, and regulate the drain valve so that the water is kept in the glass.

In operating an ordinary heater supplied with auxiliary exhaust as the heating agent, the following points should be considered:

1. Keep a steady back pressure at whatever point it is desired to operate.

2. Regulate the drain so that there is sufficient water in bottom of heater so as to form a water seal; but do not allow enough water to collect so as to cover any material amount of heating surface.

3. Keep air cock open sufficiently so that air collecting on steam side is drawn off.

4. Operate feed pump as uniformly as possible so that the highest possible average temperature may be secured.

### SAVING DUE TO USE OF FEED HEATERS.

Where the heating agent is the auxiliary exhaust from simple or compound engines and the back pressure is carried at from 15 to 25 pounds absolute, there is approximately a saving of 1 per cent. for every 10 or 11 degrees of feed heating. If a higher back pressure is carried the per cent. of saving will not be as great as this for the additional rise in temperature. But any steam used for feed heating in which its *latent heat* is really utilized to heat the feed to a higher temperature will produce some saving. *Every heat unit saved from the condenser is equal to two in the coal pile* is an axiom to consider, remember and act upon.

One of the chief devices by which heat units are rescued

from the condenser is the feed heater. The *feed heater,* the utilization of *high vacuum,* and use of *superheat* are the three greatest fuel savers that have been introduced into marine engineering practice in the last fifteen years.

A high vacuum enables you to take all possible heat out of the steam before putting it into the condenser. The feed-water heater enables you to rescue considerable of the latent heat of the steam from being lost in the condenser. Superheating enables a certain weight of steam to carry more heat than it otherwise would.

No marine plant or subdivision of the plant, is too small to justify the installation of a feed heater. When built on proper design a feed heater is almost automatic in its operation and the cost is very little compared to the saving that it will secure.

Feed heaters can be utilized in all evaporating plants, even those of small capacity, and in steam launches.

Small heaters using a copper coil can be readily installed where they will save many heat units that are otherwise wasted.

## FEED HEATERS IN EVAPORATOR PLANTS.

In the usual type of evaporating plant on merchant vessels, and also on many men of war, the evaporator is used in single effect and the vapor discharged to a distiller, where it is condensed. In such a plant all the heat in the vapor from the evaporator is lost and practically cold feed is used in the evaporator.

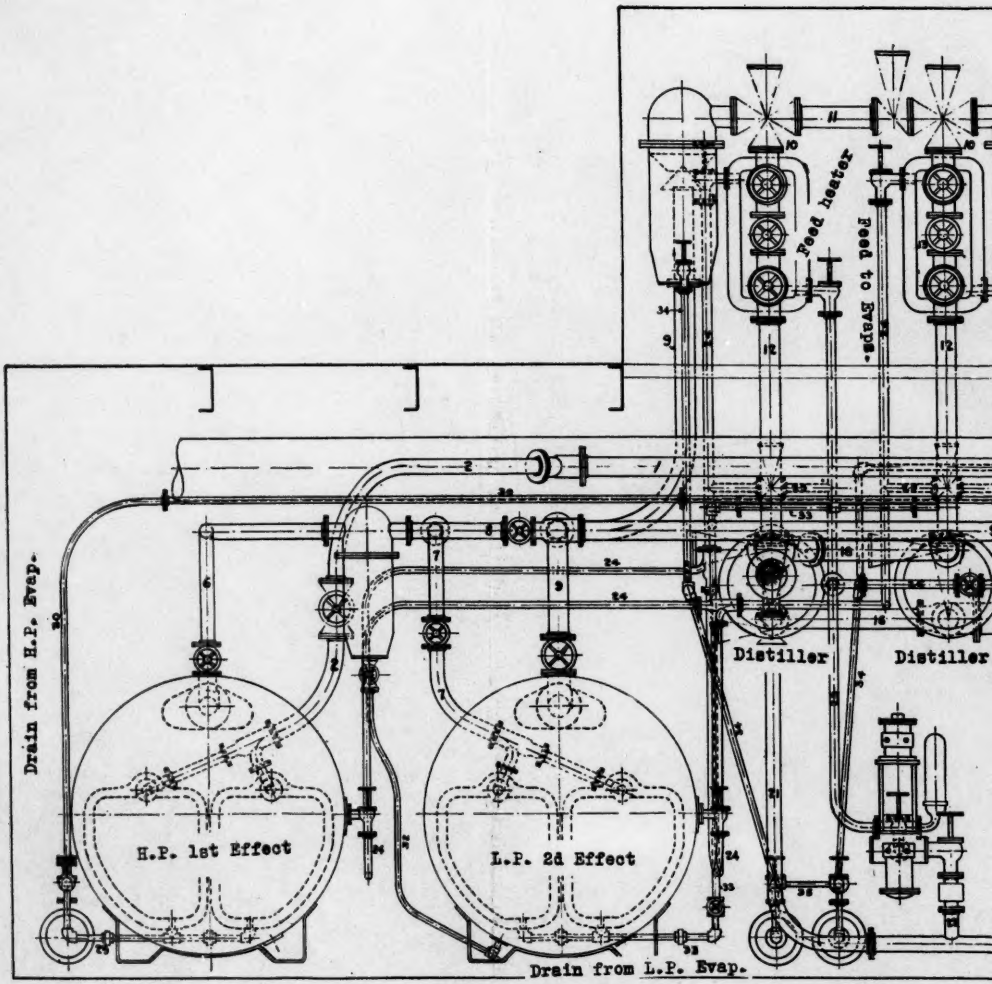
Wherever the evaporating plant is large enough to employ two evaporators, they should be piped so that they can be operated in double effect, and in any case some sort of feed-water heater should be used, and if more than two units are installed they should be arranged so that they may operate in triple or quadruple effect. Each compounding of the evaporator plant will greatly increase the economy of the plant. With single effect, a pound of steam, at about 50 pounds will produce about .8 pound of distilled water; with double effect, about 1.45 pounds; with triple effect, about 1.95 pounds. If the water

fed into the evaporator is heated by the vapor from the last effect or by means of the drains from the coils, the economy can be further increased. If the feed is raised 100 degrees (say from 100 degrees to 200 degrees), there will be an increase in economy of about 20 per cent. So that with triple-effect evaporators, using steam at 50 pounds and being provided with a feed-water heater, a pound of steam will give about 2.35 pounds of distilled water. The feed heaters used in connection with the evaporating plant may be installed in two different ways:

(1) Using the vapor from the L.P. evaporator shell for the heating agent. In this case, a small heater, about 20 feet of surface per 1,000 gallons capacity, in installed so that the vapor on leaving the L.P. shell passes through the heater and then goes to the distiller. While the feed water, after leaving the evaporator feed pump, passes through the heater on its way to the evaporators.

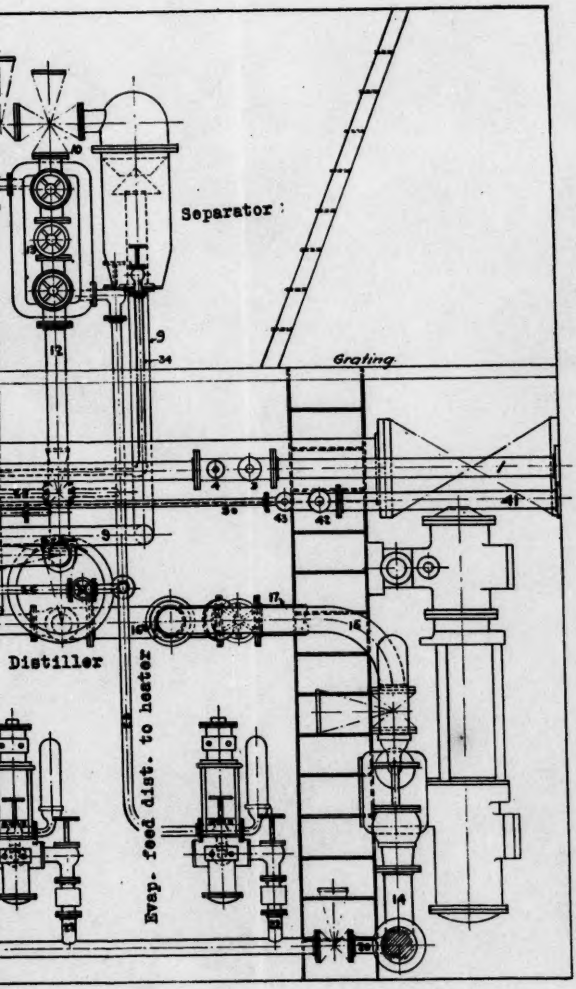
(2) Using the drain from the H.P. evaporator to heat the water. The steam in the coil of the H.P. evaporator is boiler steam, and the condensate from this is not usually potable. It is usual to drain this condensate to the feed tank or to the auxiliary condenser. This drain water, however, on leaving the coil has the same temperature as the steam in the H.P. coil. If this drain water is made to pass through a feed-water heater it may be caused to give up this heat to the feed. Hence, by a small coil-feed heater in the drain line of the H.P. evaporator the feed can be heated to a considerable extent.

The two types may be used in conjunction, first allowing the feed to pass through the vapor heater where it may be heated up to the temperature of the vapor, about 210 degrees F., and then cause the feed to the H.P. shells to pass through the H.P. coil drain heater where it can be heated to the temperature of about 270 degrees F. In a double-effect plant, with this arrangement, the feed water can enter the evaporator shells at about 210 degrees for the L.P. and 270 degrees for the H.P. where the initial pressure in the H.P. coils is 50 pounds.




*Elevation with outboard bulkhead removed looking to port si*

FIG.




| PIPE LIST | |
|---|---|
| No. | Purpose |
| 1 | Steam to evaporator room. |
| 2 | do first effect coils. |
| 3 | Steam branch to distiller circulating pump. |
| 4 | do evaporator feed pumps |
| 5 | do fresh water pumps |
| 6 | Vapor from first effect shell to first effect separator. |
| 7 | do. do. separator to second effect coils. |
| 8 | do. do. by-pass to second effect separator. |
| 9 | do. second effect shell do. do. |
| 10 | do. do. separator to feed water heater. |
| 11 | do. do. do cross connection. |
| 12 | do feed water heater to distiller. |
| 13 | do. do by-pass to distiller |
| 14 | Distiller circ pump suction from sea |
| 15 | do. disch. to combined disch. to distillers. |
| 16 | do. combined disch. to distillers |
| 17 | do. do. to fire main. |
| 18 | do. disch. from distillers to combined disch. overboard |
| 19 | do combined disch. overboard |
| 20 | Evap. feed pump suction from sea |
| 21 | do. do. dist. circ. disch. overboard. |
| 22 | do do. |
| 23 | do. disch. to feed water heater. |
| 24 | do do from feed water heater to evaporators |
| 25 | do. do. feed water heater by-pass |
| 26 | do do. cross connection |
| 27 | Fresh water pump suction from test tank. |
| 28 | do. discharge. |
| 29 | Drain from first effect coils to trap. |
| 30 | do. do. trap to aux. exhaust. |
| 31 | do do. coils by-pass trap. |
| 32 | do do. separator to shell of 2nd effect evap. |
| 33 | do. second effect coils to vapor to distillers. |
| 34 | do do. separator to trap. |
| 35 | do do. trap to evap. feed pump suction. |
| 36 | do do separator by-pass trap. |
| 37 | do distillers to test tank through water meter. |
| 38 | do do. by-pass do |
| 39 | do. distillers by-pass test tank to fresh water pump. |
| 40 | Distiller air pipe. |
| 41 | Auxiliary exhaust |
| 42 | Exhaust branch from distiller circulating pumps. |
| 43 | do. evaporator feed pumps |
| 44 | do fresh water pumps |

*g to port side.*

FIG. 7.

There is some practical objection to using temperatures above 200 degrees F. for salt water, on account of the deterioration of piping and composition fittings.

The diagrammatic arrangement of an evaporating plant using vapor feed-water heater is shown, Fig. 7. This arrangement employs traps on the H.P. coil drains. The coil-drain heaters above mentioned would be located in approximately the position occupied by the traps.

### SERIES HEATERS.

When the auxiliary exhaust is used in the feed heater the highest feed temperature that can be secured is the temperatures within a few degrees of that corresponding to the pressure in the auxiliary-exhaust line. For five pounds back pressure the temperature of the exhaust will be 227 degrees F. and the highest practical feed temperature about 222 degrees. No matter how much auxiliary exhaust may be present, this temperature cannot be exceeded unless the back pressure is increased. However, the sources of some of the heating steam, such as drains from steam pipe, H.P. and I.P. cylinders and H.P. evaporating drains, are of a higher temperature than the auxiliary exhaust. If these were allowed to heat the feed by means of separate heaters in series after leaving the auxiliary exhaust heater a higher feed temperature could be secured.

In large installations where there is abundant auxiliary exhaust for feed heating the use of such series heaters may result in securing an additional economy of several per cent.

It is generally found that at cruising speeds on naval vessels and on passenger vessels having a large number of steam-driven auxiliaries there is more auxiliary exhaust available than is required to heat the feed water to 220 degrees. Now, if a portion of the auxiliaries exhausted at a high pressure, say 40 pounds absolute, and were led to a heater into which the feed was delivered after leaving the ordinary exhaust-steam heater, operated at about 20 pounds absolute, a higher feed temperature could be obtained. If the exhaust of feed pumps

and circulating pumps at 40 pounds absolute were led to an intermediate heater a temperature of approximately 250 degrees F. might be obtained instead of the 210 degrees now secured. This would improve the economy something like 2 or 3 per cent. It would, however, entail some additional weight, and the added complication in exhaust piping and feed connections offers considerable objection.

Another arrangement would be the use of a series heater in two parts—one to use the auxiliary exhaust at about 20 pounds absolute, giving a feed temperature of 210 degrees F., and a high-pressure heater, fed with steam from the first receiver of the engine, the pressure being approximately 100 pounds absolute. With 10 per cent. of engine steam used for this H.P. heater, a temperature of about 350 degrees F. should be obtained, resulting in an increase in economy of 3 to 4 per cent. The increase in weight for this installation above the ordinary exhaust heater would not be more than a few tons on a battleship, a weight made up in a few days' steaming. There is, however, the increase in steam and feed connections which would offer considerable objections.

If steam of boiler pressure is used instead of that from the first receiver, a still higher temperature of feed can be obtained, but more trouble with the heater is likely to be encountered and it will have to be considerably heavier.

Another simpler and more feasible proposition is to use heating coils in the delivery chamber of the exhaust-steam heater; the coils to be supplied with steam from (a) evaporator coils, (b) I.P. receiver, (c) drains from separators and steam pipes. Such coils may be located in the discharge chamber of the heater, with very little additional space and very little extra weight or piping. Such an arrangement is indicated in sketch. The heater is designed like an ordinary single-pass closed heater. The discharge chamber is made slightly deeper, and in it are placed a series of coils. The first series might take their steam and condensed water from the evaporator-coil drain, the pressure of this being between 60 and 80 pounds absolute, with corresponding temperature of 293 to 312 degrees F. The

second series will take their steam from the I.P. receiver, the pressure here varying from 50 to 115 pounds (depending on speed), and corresponding temperature 280 to 338 degrees F. The third series will take steam from the steamline separator

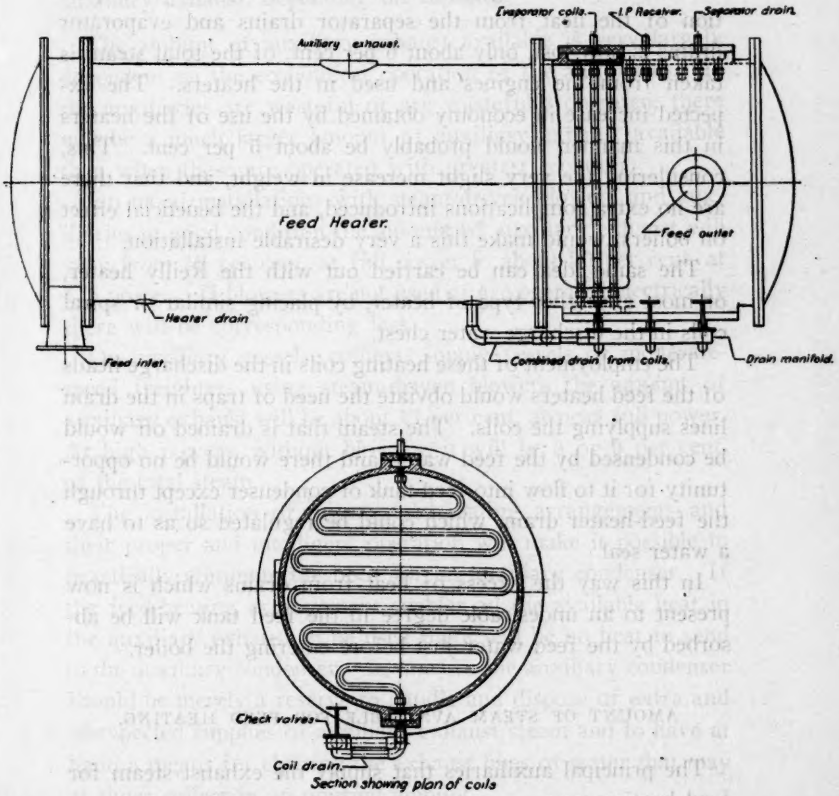



FIG. 8.—FEED HEATER FITTED WITH EXTRA HEATING COILS IN DISCHARGE CHAMBER.

drains, the pressure being 200 to 300 pounds (depending on speed), and corresponding temperature 381 to 417 degrees F. By means of these coils, using 2 to 3 per cent. of the total steam from the evaporator-drain coils, heating water approximately

to 235 to 245 degrees F.; using 5 per cent. of total steam in I.P. receiver coils, heating water to approximately 280 to 320 degrees F.; using 3 per cent. of total steam in separator-drain coils at boiler pressure, a temperature of 340 to 360 degrees F. for feed water might be secured. Since a considerable portion of the heat from the separator drains and evaporator drains is now lost, only about 6 per cent. of the total steam is taken from the engines and used in the heaters. The expected increase in economy obtained by the use of the heaters in this manner would probably be about 5 per cent. This, considering the very slight increase in weight, and that there are no extra complications introduced, and the beneficial effect on boilers, would make this a very desirable installation.

The same idea can be carried out with the Reilly heater, or most any other type of heater, by placing similar or spiral coils in the discharge water chest.

The employment of these heating coils in the discharge heads of the feed heaters would obviate the need of traps in the drain lines supplying the coils. The steam that is drained off would be condensed by the feed water, and there would be no opportunity for it to flow into feed tank or condenser except through the feed-heater drain, which could be regulated so as to have a water seal.

In this way the excess of heat from drains which is now present to an undesirable degree in the feed tank will be absorbed by the feed water just before entering the boiler.

## AMOUNT OF STEAM AVAILABLE FOR FEED HEATING.

The principal auxiliaries that supply the exhaust steam for feed heating are:

| | Per cent. of total steam. |
|---|---|
| Feed pumps | use 3.0 to 5.0 |
| Circulating pumps | use 1.5 to 2.5 |
| Air pumps | use 0.2 to 0.5 |
| Forced-draft blowers | use 1.5 to 2.5. |

| | Per cent. of total steam. |
|---|---|
| Bilge pumps, ice machines, steering engine and other miscellaneous machinery..... | use 1.0 to 5.0 |
| Dynamo plant, other than for F.D. blowers. | use 1.0 to 2.0 |
| Evaporating plant, when discharging vapor to auxiliary exhaust, depending on capacity. | |

The amount of auxiliary exhaust available is very largely dependent on the economical operation of the auxiliaries. If the auxiliaries are wasteful or are wastefully operated, there will be a much larger amount of auxiliary exhaust available than when these are operated with greatest economy.

On naval installations with steam-driven blowers and auxiliaries in good condition the amount of auxiliary exhaust will vary from 12 per cent. at full power to about 20 per cent. at 1/5 power. If blowers are not used or are operated electrically there will be corresponding less.

On auxiliary vessels, colliers, supply vessels or moderate-speed freighters using steam-driven blowers the amount of auxiliary exhaust will be about 11 per cent. at near full power. At high powers, without blowers, it will be 8 or 9 per cent. of the total steam.

The installation of proper feed-heating arrangements and their proper and intelligent operation will make it possible to practically eliminate the need for an auxiliary condenser. If the feed-heating arrangement enables all the available heat in the auxiliary exhaust to be used there will be no heat to send to the auxiliary condenser. In practice the auxiliary condenser should be merely a reserve to handle and dispose of extra and unexpected supplies of auxiliary exhaust steam and to have at hand a means for clearing the exhaust lines of water that may at times collect in an unusual amount.

## TYPES OF HEATERS.

A number of different special types of heaters that are in successful use for marine service will be described and illustrated. Information concerning the operation and construc-

tion of these heaters has to a large extent been obtained from the makers of the apparatus. The comments made upon their design, construction and operation are based on the results of actual experience in operation.

The types of heaters described are those most largely used and which have to a considerable extent been standardized.

*The Normand Feed-Water Heater.*—The Normand feed-water heater is a type developed by Normand, of Havre, France. It has been built for marine service in this country by the Bath Iron Works. This design of heater is perhaps the most efficient of the plain-tube heaters. In its design and construction correct basic principles are applied, and this, with good workmanship, is largely accountable for its success.

The heater is a dead-end heater and is usually installed in a vertical position. The tubes are straight and are expanded into the tube sheets. Whirling plates are fitted in the tubes. The water chamber is provided with a scattering plate which holds these whirling plates in place.

The exhaust steam is allowed to enter near the top and near the feed exit. The steam is baffled so as to pass back and forth across the tubes. A drain is fitted near the bottom of shell. About 18 inches from the bottom of heater there is an air vent. This air vent leads by means of a small pipe through a coil in the receiving-water chamber and is then led to condenser or atmosphere. This air vent draws off the air that collects above the water. As this air is likely to contain considerable vapor and heat, it is led through the coils in the receiving-water chamber, and thus the heat contained in the air that is drawn off is utilized to heat the feed water. By opening this air vent the air, as it accumulates, is continuously being drawn off and does not have a chance to collect and render the heating surface inactive.

The details of this heater can be seen from Figure 9. Expansion is allowed in the attachment of shell to end flanges.

This type of heater is not expensive to build, is very durable and efficient. It also can be readily repaired without the need of carrying special parts.

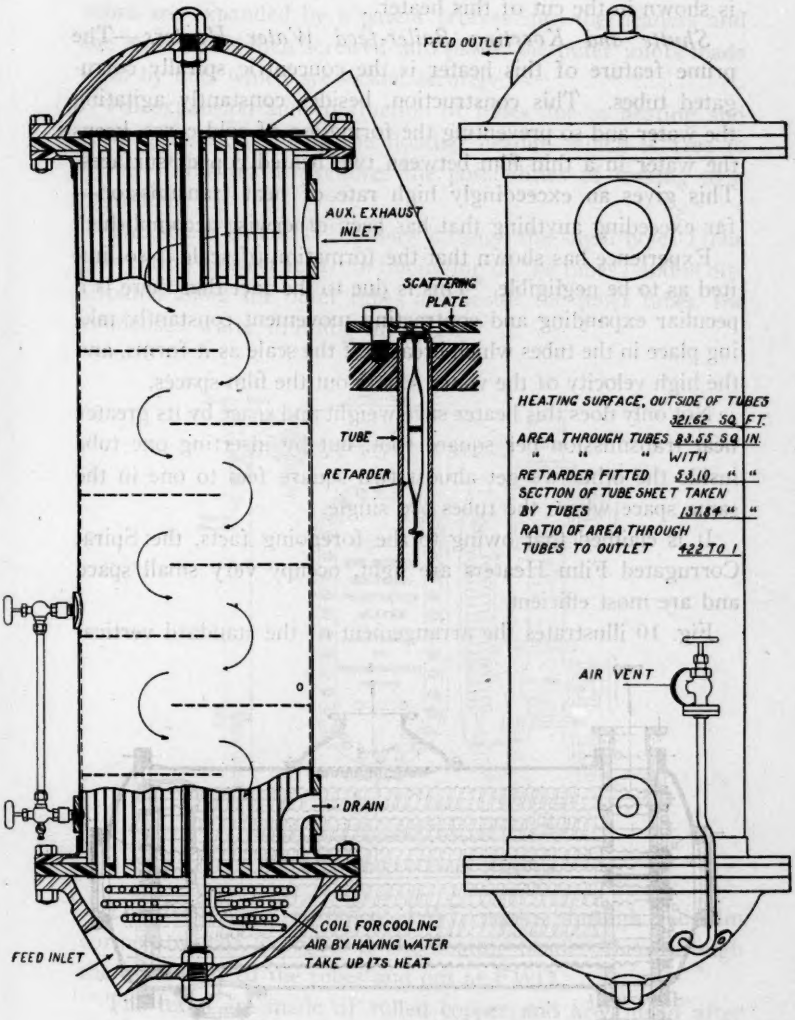

FEED OUTLET
AUX. EXHAUST INLET
SCATTERING PLATE
TUBE
RETARDER
DRAIN
COIL FOR COOLING AIR BY HAVING WATER TAKE UP ITS HEAT
FEED INLET
HEATING SURFACE, OUTSIDE OF TUBES 321.62 SQ FT.
AREA THROUGH TUBES 83.55 SQ IN.
" " " WITH
RETARDER FITTED 53.10 " "
SECTION OF TUBE SHEET TAKEN
BY TUBES 137.84 " "
RATIO OF AREA THROUGH
TUBES TO OUTLET 422 TO 1
AIR VENT

FIG. 9.—NORMAND FEED HEATER.

A method of efficiently staying tube sheets and end castings is shown in the cut of this heater.

*Shutte and Koerting Boiler-feed Water Heaters.*—The prime feature of this heater is the concentric spirally corrugated tubes. This construction, besides constantly agitating the water and so preventing the formation of cold cores, keeps the water in a thin film between two heated copper surfaces. This gives an exceedingly high rate of heat transmission—far exceeding anything that has been otherwise accomplished.

Experience has shown that the formation of scale is so limited as to be negligible. This is due to the fact that there is a peculiar expanding and contracting movement constantly taking place in the tubes which breaks off the scale as it forms, and the high velocity of the water scours out the film spaces.

Not only does this heater save weight and space by its greater heat transmission per square foot, but by inserting one tube inside the other we get almost two square feet to one in the same space where the tubes are single.

It is claimed that owing to the foregoing facts, the Spiral Corrugated Film Heaters are light, occupy very small space and are most efficient.

Fig. 10 illustrates the arrangement of the standard vertical

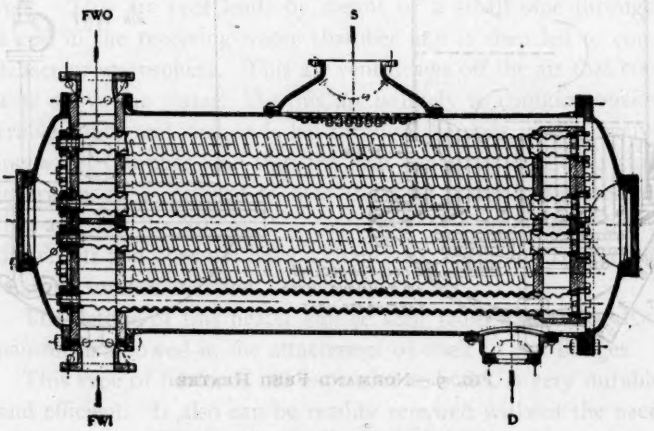



FIG. 10.—S. AND K. FEED HEATER.

and horizontal types of boiler-feed-water heaters. The outer tubes are expanded by a patent process into the headers and the inner tubes then screwed into place; the outer joints made tight by nuts which are locked securely.

All expansion and contraction of tubes due to heating and cooling is taken up by the floating header, which is free to move. This feature reduces the possibilities of leaking to a minimum.

The steam enters at S passes through the small holes (thus preventing a direct jet from impinging on the tubes) and is distributed about the outer tubes and inside the inner tubes, the condensed water draining off at D.

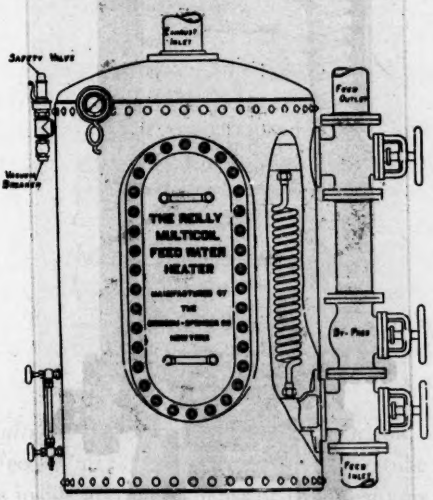



FIG. 11.—REILLY FEED HEATER.

The feed water enters at FWI, passes through the film formed by half the tubes to the floating header, then through the other half of the tubes and out at FWO.

The tubes are made of rolled copper and are tinned after being corrugated.

This type of heater is very efficient and is comparatively light and compact. It has the disadvantage of producing con-

siderable resistance to the flow of water through the tubes. In using them near their full capacity there is a considerable drop in pressure from inlet to outlet. For this reason there should be a fairly good margin of capacity.

FIG. 12.

This heater makes use of the basic principles of heating to a very great extent. The water is in a thin film and is given a whirling movement at relatively high velocity. The exhaust

steam enters near the feed outlet, and the path of the steam is nearly at right angles to the tubes.

This heater is a comparatively new development, and as yet has not been very extensively used in commercial marine work. Difficulty has been experienced in keeping gaskets at tube ends tight, and after a certain amount of service the corrugated tubes have been known to crack.

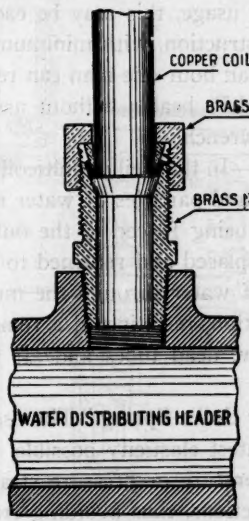



FIG. 13

*Reilly Multicoil Heater.*—General Construction. The Reilly Multicoil Feed-Water Heater is of the water-tube type, *i. e.*, the water is inside and the exhaust or live steam on the outside of the tubes. The heating surface consists of a number of helical coils of double-refined seamless-drawn copper tubing of small diameter. This construction makes the heating surface highly efficient and exceptionally light and compact. The connections between the coils and headers are made with patented screwed union joints without brazing.

Tube Fastenings.—The coils are connected to the top and bottom distributing headers by screwed union joints, and every

coil and union is standard and interchangeable—an important point under some conditions of service. Fig. 13.

Easy Accessibility.—All Reilly heaters are furnished with a large door in the front of the shell, which permits of free access to all internal parts and connections without the inconvenient necessity of disconnecting the heater supports.

Should it be necessary to inspect or repair the heater, owing to accident or wrong usage, this may be easily accomplished, due to the above construction, with minimum trouble, expense and loss of time. In an hour one man can remove and replace every coil in a 750-H.P. heater without assistance and with no other tool than a wrench.

Active Circulation.—In the Reilly multicoil heater the centrifugal action is great, and particles of water near the center of the tubes are always being forced to the outside, while those on the outside are displaced and returned to the center. This rapid swirling flow of water through the multicoils insures a constant violent agitation, resulting in a remarkably high efficiency. Sluggish flow, dead tubes and air pockets are positively avoided.

Elasticity.—"Every Coil a Spring." The use of coiled copper tubes insures the greatest elasticity possible, and prevents the transmission of temperature or pressure strains to the heater shell or its pipe connections, thus avoiding cracked coils.

Back Pressure.—The unusually large space around the tubes for the passage of steam through the shell of the heater (largely in excess of the cross-sectional area of the exhaust pipe) minimizes the resistance due to the passage of the exhaust steam, and so eliminates the possibility of additional back pressure on the engine.

This heater has no baffling for the steam, but the proper conditions with regard to counter-current principle and a flow of steam at nearly right angles to the flow of water are secured by the path of steam downward across the coils. This heater is very largely used in marine work and is generally satisfactory.

*The Wainwright Heater.*—This type of heater employs spirally-corrugated tubes, expanded securely into the tube sheets. The corrugated tube makes provision for expansion. In this heater the tubes are brass thimbled at their ends and the thimble, which is placed within the tube, is expanded with the tube by means of a special expander. Fig 16.

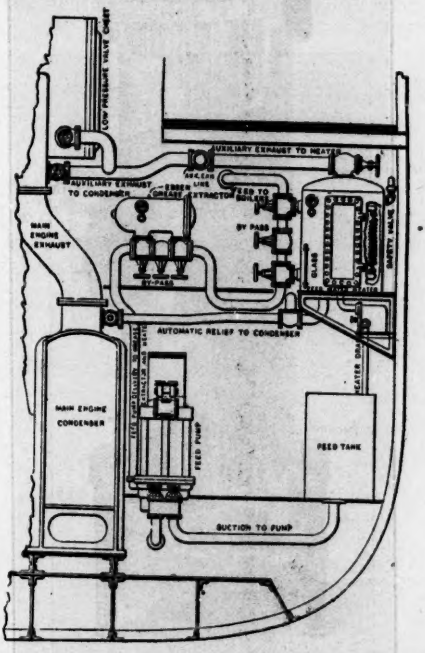



FIG. 14.—HEATER CONNECTION, IN CONNECTION WITH REILLY HEATER.

The tubes being corrugated, the water is agitated and given a spiral movement which causes a whirling movement of the particles, thus bringing more of them in contact with the heating surface and causing better transmission of heat.

The tubes are divided into several groups, and the water is sent back and forth several times through the heater in a direction opposite to the flow of steam. This arrangement also allows the exhaust openings to be placed so that the cold water

entering meets the outgoing steam, while the heated water, just as it leaves the heater, receives the full benefit of the heat of entering exhaust.

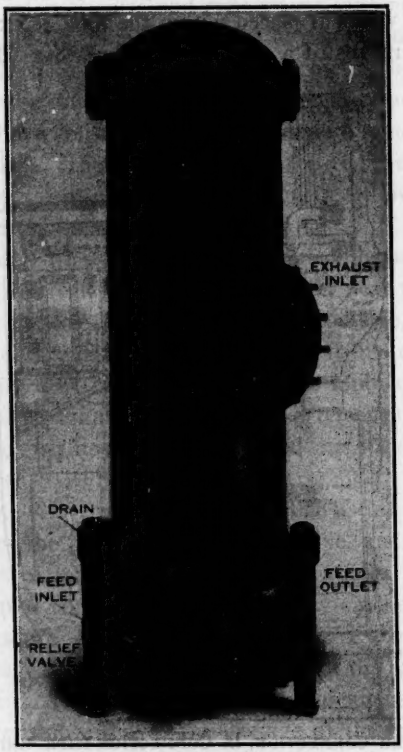



FIG. 15.—WAINWRIGHT FEED HEATER.

Although the velocity of flow is greater than in ordinary heaters, it does not approach a velocity that will produce objectionable friction. In the design of heater this point is given consideration, and sufficient velocity is given to secure efficient heat transmission without exceeding a reasonable amount of friction.

In other types of Wainwright heaters the steam is not

baffled, but the exhaust inlet is placed at the center of the steam space on the side of the feed outlet, since by so placing it the length of the path of the steam is the shortest and the resistance to its flow is reduced to a minimum. The spaces between the nests of tubes caused by the partition plates in the water spaces make unobstructed paths for the steam to reach any part of the heater without air binding.

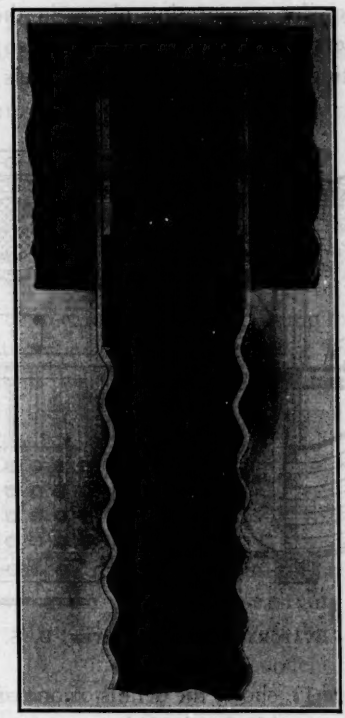


FIG. 16.—METHOD OF EXPANDING TUBES OF WAINWRIGHT FEED HEATER.

The fact that this make of heater employs in one case no guide baffles for the steam and in another case causes the steam to pass among the tubes in a direction opposite to the direction

of the flow of water, is a noticeable fact. It is believed that the counter-current principle is the correct one, but the flow of steam should be across and not along the tubes, in order to secure best efficiency. This fact has been amply proved in experiments with baffling water-tube boilers.

*Bureau of Steam Engineering Curved-Type Heater.*—A novel type of curved-tube heater is that designed by the Bureau of Steam Engineering, U. S. Navy, and very extensively used as a feed heater for evaporating plants in the U. S. Naval Service. Heaters of the same type have also been built for boiler-feed heating on some of the large vessels of the navy.

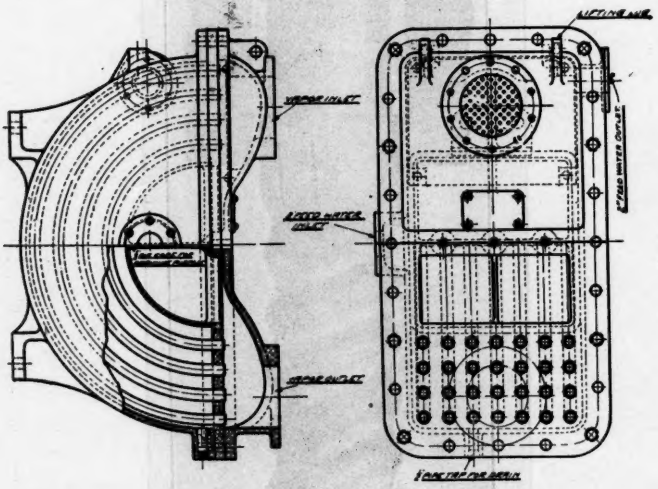


FIG. 17.—EVAPORATOR FEED HEATER, U. S. NAVY.

The cut, Figure 17, shows the details of one of these heaters. The tubes are composition and are bent to the curvature of a semicircle, and the ends are expanded into the composition tube sheets. The steam in this case, when used for an evaporator-feed heater, is on the inside of the tubes and the water on the outside. In this type of heater, when used for boiler-feed heating, the steam is on the outside of the tubes and the water

on the inside. Baffles also are provided for directing the steam across the tubes. The curvature of the tubes allows for any possible expansion and also causes the water to receive a rolling motion as it passes through the tubes. It is, however, believed that this rolling motion thus secured is not as efficient for heating as that secured from spiral tubes or spiral films.

In this type of heater any danger of leakage from the salt water to the vapor side is well guarded against, and this is a most important point in any heater used for evaporator-feed heating in which case there is salt water to be heated.

## EFFECT OF CONDENSER ARRANGEMENTS AND OPERATION ON FEED HEATING.

To secure the greatest economy the air-pump discharge should be as high as possible consistent with the vacuum carried. The temperature of the water leaving the condenser is dependent upon the amount and temperature of the circulating water. It is, of course, uneconomical to pump more circulating water through the condenser than sufficient to condense the steam and maintain the vacuum. In the design of condensers with reference to the path of the circulating water the following should be aimed at:

(1) Require the least possible circulating water.

(2) Have the condensed water at the highest possible temperature consistent with the vacuum carried.

To secure the above results the steam on entering the condenser should strike the colder water-carrying tubes and the condensed water should be in contact with the warmest water-carrying tubes. In ordinary marine condensers the exhaust steam usually enters at the top of the condenser, and in such cases the cooling water should pass first through the top tubes and the entrance of cooling water should be as near as possible to the exhaust connection.

By doing this the cooling water acts on the steam first where there is the greatest difference in temperature, hence the greatest condensing effect. As the water flows through the con-

denser it is gradually raised in temperature, and if just the right amount is admitted it, it would leave at a temperature that would correspond to the temperature of the vapor. With the ideal arrangement the cooling water would absorb only the amount of heat necessary to cause condensation and would leave the water, as it goes to the air pump suction, at the same temperature as the vapor in the condenser. Since every heat unit taken away from that given to the circulating water is a gain, it should be the aim of condenser design to give the discharge water and the air-pump suction the temperature corresponding to the vapor pressure in the condenser. If too much circulating water is supplied the water in the condenser will be cooled beyond this point and the corresponding amount of heat lost in the circulating water, while if too little circulating water is supplied the vapor pressure will rise and the engine will not have as good a vacuum.

If the circulating water is allowed to enter at a point away from the exhaust it will cool the water in the bottom of the condenser more than necessary and will be considerably heated by the time it comes in contact with the steam near the exhaust entrance. Hence, it is taking heat away from the water already condensed and is thereby rendered less able to condense the actual exhaust steam. In order to obtain the same effective vacuum an excess of water is pumped through the condenser and the air-pump discharge is cooled below the temperature necessary to secure this vacuum. As a consequence the water arrives at the feed tank at a temperature considerably lower than that necessary to secure the vacuum and, with the same amount of auxiliary exhaust available in the feed heater, a lower feed temperature will be secured than if the flow of circulating water had been in the opposite direction, and the water on leaving the condenser would have the temperature corresponding to the vacuum carried.

If the air-pump discharge with a 28-inch vacuum is 100 degrees F. and if there is 10 per cent. of the total steam available in the auxiliary exhaust at 20 pounds absolute, and 5 per

cent. of the total steam available (from evaporator-coil drains, steam lines and receiver drains, etc.), for use in a secondary heater at a pressure of 38 pounds absolute, a feed temperature of 260 degrees F. can be secured. On the other hand, if the drains, etc., which are at a higher pressure than the auxiliary exhaust, are led to the same feed heater that uses the auxiliary exhaust only 227 degrees F. feed temperature is possible.

If, instead of having the air-pump discharge at 100 degrees, this is reduced to 80 degrees F. by bad condenser design, or by running the circulating pump too fast, with the same amount of exhaust as above, only 230 degrees F. can be obtained instead of 260 degrees F.

From this it will be seen that the design and the operation of the condensing plant in the matter of circulating water can make as much as 3 or 4 per cent. difference in economy, while the fitting of a series feed-water heater may make about 4 per cent. difference in economy. These are relatively small savings but when added together make a very considerable difference in the amount of fuel burnt.

# U. S. S. *AYLWIN, PARKER* AND *BENHAM.*

## CONTRACT TRIAL PERFORMANCE.

By John S. McKinney, Associate.

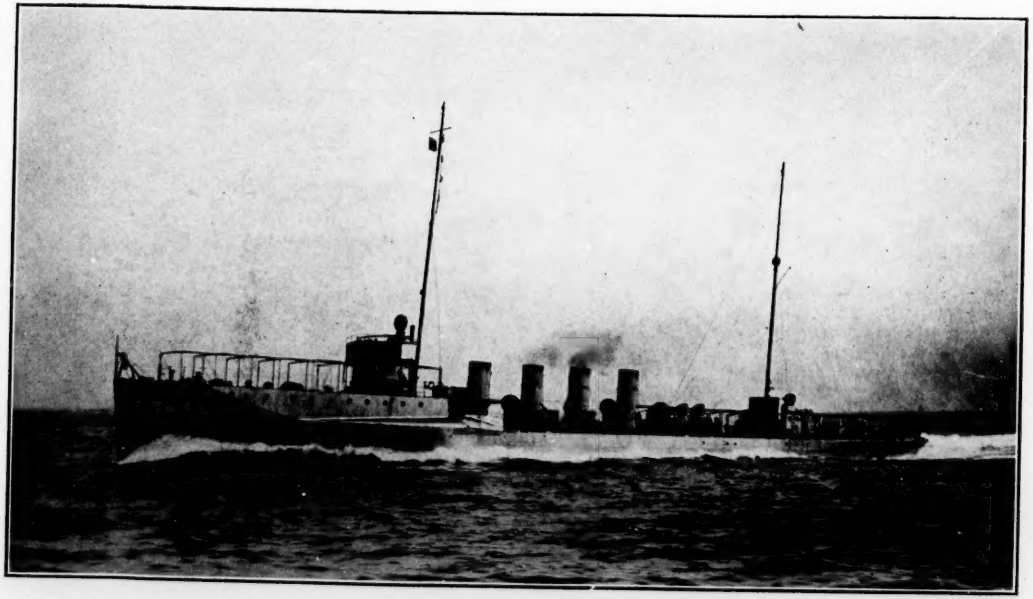
The construction of eight torpedo-boat destroyers, Nos. 43 to 50, inclusive, was authorized by an Act of Congress approved March 4, 1911, of which the *Aylwin* is No. 47, the *Parker* is No. 48, and the *Benham* is No. 49.

These vessels were built under contract by the William Cramp and Sons Ship and Engine Building Company of Philadelphia, Pa., the contract speed being 29.5 knots, the trial displacement 1,036 tons and the price of each vessel $756,100. The time of completion for the *Aylwin* was twenty-two and one-half months; for the *Parker,* twenty-three months, and for the *Benham,* twenty-three and one-half months from the date of contract.

These are twin-screw vessels with combinations of turbines and reciprocating engines. The turbines are of the Cramp-Zoelly type, to be used when speeds greater than 17 knots are desired. For cruising speeds below 17 knots there is on each shaft a reciprocating engine.

### HULL.

| Principal dimensions: | |
|---|---|
| Length on L.W.L., feet and inches | 300–00 |
| over all, feet and inches | 305–03 |
| Breadth, extreme at L.W.L., feet and inches | 30–04 |
| Ratio of length to beam | 10.064 |
| Draught to L.W.L., feet and inches | 9–02½ |
| Displacement corresponding, tons | 1,010 |
| per inch at L.W.L., tons | 14.21 |
| Area immersed midship section, square feet | 190.0 |
| Coefficient of fineness, block | 0.415 |



Photograph By DAWSON, Phila., Pa.

U. S. S. "BENHAM."—FULL-SPEED STANDARDIZATION TRIAL, DEC. 16, 1913.

The hull is of steel, galvanized in all parts below the water line. The vessel is of the raised-forecastle type, which is carried aft to frame No. 46. On the forecastle deck are located the anchors, capstan and pilot house, with flying bridge. A searchlight is carried on a raised platform above the bridge.

Aft of frame No. 46 the main deck is a weather deck. From forward aft on this deck are located storeroom, lamp room, wardroom officers' quarters, wardroom, pantry and galley. Aft, in a deck house, are located hatch trunk to crew's quarters, wireless room and crew's water closets. A searchlight is carried on top of this house. There are two masts equipped with wireless outfit and signal yards.

The berth deck is divided into two parts. It extends from the stem to frame No. 48 and from frame No. 134 aft, the space between being used for deep fuel-oil tanks, cofferdam, firerooms and engine rooms. Forward are located a storeroom, paint and oil room and crew's quarters, and aft are crew's and petty officer's quarters and storerooms.

In the hold are located trimming tanks, chain lockers, storerooms, ammunition and handling rooms, fuel-oil tanks, cofferdam (extending to the main deck and designed to be used as a reserve-feed tank), boiler and engine rooms, oil tanks, ammunition and handling rooms and after trimming tanks. Reserve-feed and fresh-water tanks are located abreast boilers No. 2 and 3. An engineer's storeroom is carried over the fresh-water tank on the starboard side and over the reserve-feed tank on the port side. Coal for galley use is carried in a small bunker located in the port forward corner of No. 1 fireroom.

BATTERY.

Four 4-inch R.F. guns.

Four 5.2-m. by 45-cm. twin torpedo tubes.

The 4-inch R.F. guns are located as follows: One on the forecastle-deck center line, commanding ahead and broadside fire; one starboard and one port, at the break of the forecastle; and one well aft on the main-deck center line.

The torpedo tubes are located on the main deck. One is on the port side above frame No. 74 and one on the starboard side over frame No. 95. The remaining two are abreast one another over frame No. 116.

### BOATS.

On the starboard side about midships are carried a 20-foot whale boat and a 21-foot motor dory, swung in davits. On the port side are a 20-foot whale boat and a 14-foot wherry swung in davits.

### ANCHOR WINDLASS.

The anchor windlass is located on the forecastle deck at frame No. 11. It is of the vertical type, made by the American Engineering Company, of Philadelphia, Pa. The engine is located below on the after bulkhead of the lamp room. The principal dimensions are as follows:

| | |
|---|---|
| Number of cylinders | 2 |
| Diameter of cylinders, each, inches | 5 |
| Stroke, inches | 5 |

### STEERING ENGINE AND GEAR.

The ship is equipped with three steering stations: (1) top of pilot house, steam gear only; (2) pilot house, combined steam and hand gear; (3) hand gear on main deck aft. The steering engine is in the pilot house and is of the following dimensions:

| | |
|---|---|
| Number of cylinders | 2 |
| Diameter of cylinders, inches | 6 |
| Stroke, inches | 6 |

### FIRE MAIN.

The fire main is of steel pipe, lead lined, of 3-inch outside diameter, and extends throughout the machinery space directly under the main-deck beams. The main is provided with a cut-out valve dividing it into a forward and after system.

It is directly connected through reducing valves to the flushing system and connections are also provided for flushing the stern tubes and oil coolers in case of emergency.

### FLUSHING SYSTEM.

The flushing system is taken direct from the fire main and is designed to supply water to the officers' bath, galley and pantry forward and to the crew's water closets and washroom aft.

### FRESH-WATER SYSTEM.

Fresh water is carried in two tanks, starboard and port, of about 4,000 gallons combined capacity, located in the hold of the ship between frames Nos. 63 and 74. The tanks are provided with filling connections upon each side of the vessel fitted with hose connections. There is also a connection to the distilling plant. A small gravity tank of about 50 gallons capacity is located just aft the pilot house on the forcastle deck, supplying the galley and officers' showers.

### DRAINAGE SYSTEM.

The main extends from the after end of the auxiliary room to the forward fireroom and is connected to the boiler, engine and auxiliary rooms, branches to the bilges being fitted with stop-check valves and strainers; where ends of suction pipes are inaccessible strainers are not fitted. Connections are provided to drain compartments below the berth deck forward of and abaft the machinery spaces.

### VENTILATION.

Living spaces throughout the vessel are ventilated by natural means, boiler rooms being ventilated by means of forced-draft blowers.

## HEATING.

The vessel is equipped with the usual system of brass-pipe coil radiators. The steam for this service in the living spaces forward is taken from the auxiliary steam line in the forward fireroom and for the quarters aft is taken from the auxiliary steam line in the engine room.

## MAIN ENGINES.

The propelling units are two Cramp-Zoelley turbines and two compound reciprocating engines.

The turbines are of the compound impulse type arranged on two shafts and designed to develop 16,000 S.H.P. at about 620 revolutions per minute. The reciprocating engines, two in number, are of the vertical, inverted, two-cylinder, compound type located just forward of the turbines. These are brought into use by means of a power-controlled clutch of the friction type. This clutch is so designed that the engines are capable of being connected or disconnected without slowing down and is interlocked with the maneuvering valves so that steam can not be admitted to the ahead or astern turbines until the reciprocating engines are disconnected. There is also fitted a governor valve, which secures the reciprocating engine from running away in case the clutch should become disconnected.

The turbines consist of two essential parts: the fixed part or casing and the moving part or rotor. Within the casing are secured the fixed diaphragm plates, the intermediate-guide blades and the expanding-guide blades, and on the rotor are secured the discs and drum carrying the moving blades. The rotor is supported at the forward and after ends of the casing by bearings secured thereto.

The ahead and astern turbines, with the exhaust belt between them, are within one casing of cast iron. This casing is divided along its horizontal axis so that its upper halves of the ahead, astern and exhaust belts may be removed in one operation for purposes of inspection or repair.

## — TURBINE DATA —

| STAGE. | AHEAD. | | | | | | | | | | | | | | | | ASTERN. | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | | | | DRUM | | | | | | | | | DRUM. | | | | |
| | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 |
| PITCH DIAMETER, INCHES | 63 | 63 | 63 | 63 | 63 | 63 | 63 | 63 | 63 | 63 | 63 | 63 | 63 | 64 13/32 | 65 1/8 | 67 11/16 | 55 1/8 | 55 1/8 | 55 1/8 | 55 1/8 | 55 1/8 | 55 1/8 | 55 1/8 | 55 1/8 |
| ARC OF GUIDE-BLADE OPENINGS IN DIAPHRAGM, DEGREES | 135 | 145 | 315 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | 360 | — | — | — | — | — | 295 | 360 | 360 | 360 | — | — | — | — |
| RADIAL HEIGHT OF GUIDE BLADE OPENINGS IN DIAPHRAGM, INCHES | 1/8 | 13/64 | 11/64 | 3/16 | 13/64 | 17/64 | 15/64 | 11/64 | 1 5/64 | 1 13/64 | 1 11/64 | — | — | — | — | — | 13/64 | 3/16 | 23/64 | 1 3/64 | — | — | — | — |
| NUMBER OF ROWS OF ROTOR BLADES | 2 | 2 | 2 | 2 | 2 | 2 | 2 | 2 | 2 | 2 | 2 | 3 | 3 | 3 | 3 | 6 | 2 | 2 | 2 | 2 | 2 | 2 | 2 | 2 |
| " " " " CYLINDER BLADES | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 3 | 3 | 3 | 3 | 6 | 1 | 1 | 1 | 1 | 2 | 2 | 2 | 2 |
| HEIGHT OF BLADES, CYLINDER, INCHES | — | — | — | — | — | — | — | — | — | — | — | 3 15/32 | 4 11/32 | 6 1/8 | 6 11/16 | 8 23/64 | — | — | — | — | 1 23/64 | 3 7/8 | 5 15/64 | 5 11/64 |
| " " " ROTOR " | 9/32 | 5/16 | 23/64 | 11/32 | 5/16 | 13/32 | 23/64 | 1 5/16 | 1 13/64 | 2 7/64 | 2 25/64 | " | " | " | " | " | 23/64 | 17/32 | 23/64 | 1 19/64 | " | " | " | " |
| " " " CYLINDER " | 23/64 | 15/32 | 13/32 | 27/64 | 1/2 | 31/64 | 1 3/64 | 1 5/16 | 1 13/32 | 2 15/64 | 3 5/16 | " | " | " | " | " | 23/64 | 15/32 | 1 3/32 | 1 21/64 | " | " | " | " |
| " " " ROTOR " | 13/32 | 1 7/64 | 1 11/32 | 13/32 | 1 3/64 | 1 3/8 | 1 1/2 | 1 11/16 | 2 1/16 | 2 21/64 | 3 13/32 | " | " | " | " | " | 43/64 | 17/32 | 1 3/8 | 1 27/64 | " | " | " | " |
| " " " CYLINDER " | — | — | — | — | — | — | — | — | — | — | — | " | " | " | " | " | — | — | — | — | — | — | — | — |
| " " " ROTOR " | — | — | — | — | — | — | — | — | — | — | — | " | " | " | " | " | — | — | — | — | — | — | — | — |
| " " " CYLINDER " | — | — | — | — | — | — | — | — | — | — | — | — | — | — | — | " | — | — | — | — | — | — | — | — |
| " " " ROTOR " | — | — | — | — | — | — | — | — | — | — | — | — | — | — | — | " | — | — | — | — | — | — | — | — |
| " " " CYLINDER " | — | — | — | — | — | — | — | — | — | — | — | — | — | — | — | " | — | — | — | — | — | — | — | — |
| " " " ROTOR " | — | — | — | — | — | — | — | — | — | — | — | — | — | — | — | " | — | — | — | — | — | — | — | — |
| " " " CYLINDER " | — | — | — | — | — | — | — | — | — | — | — | — | — | — | — | " | — | — | — | — | — | — | — | — |
| " " " ROTOR " | — | — | — | — | — | — | — | — | — | — | — | — | — | — | — | " | — | — | — | — | — | — | — | — |

## — CRUISING ENGINE DATA. —

TYPE ——— VERTICAL — INVERTED — TWO CYLINDER — COMPOUND.

| | H.P. | L.P. |
|---|---|---|
| CYLINDER DIAMETER, INCHES | 13 | 25 |
| STROKE, INCHES | 12 | 12 |
| DIAMETER PISTON ROD, INCHES | 3 | 3 |
| CYLINDER CLEARANCES, TOP, LINEAR | 1/4 | 1/4 |
| " , PER CENT | 14.8 | 13.4 |
| " , CUBIC INCHES | 236 | 788 |
| BOTTOM, LINEAR | 5/16 | 5/16 |
| " , PER CENT | 10.8 | 10.9 |
| " , CUBIC INCHES | 173 | 642 |

| | H.P. | L.P. |
|---|---|---|
| STEAM LEAD, TOP, STARBOARD, INCHES (MEASURED) | 3/32 | 11/32 |
| " , PORT, INCHES " | 9/32 | 9/32 |
| BOTTOM, STARBOARD, INCHES " | 15/32 | 7/16 |
| " , PORT, INCHES " | 13/32 | 3/8 |
| EXHAUST LEAD, TOP, STARBOARD, INCHES " | 1 5/16 | 1 9/32 |
| " , PORT, INCHES " | 1 17/64 | 1 17/64 |
| BOTTOM, STARBOARD, INCHES " | 61/64 | 1 |
| " , PORT, INCHES " | 61/64 | 51/32 |

TABLE I.

U.S.S. PARKER

TURBINE AND CRUISING ENGINE DATA.

*Full Speed Ahead.*—Steam at full pressure is admitted to the full-head chest and allowed to expand through all the stages of the turbines until finally exhausted into the condensers. The astern turbines under these conditions are running idly in a vacuum.

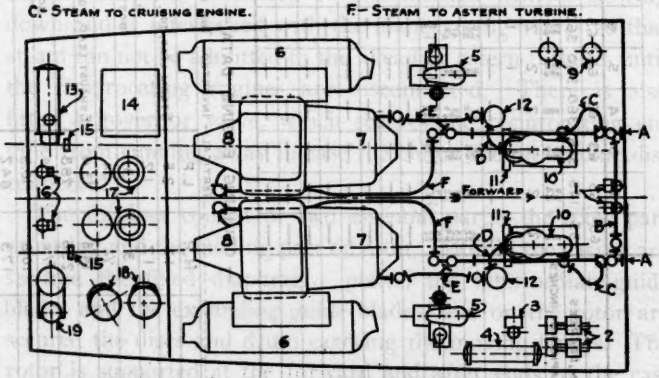
*High Cruising Speeds.*—At these speeds the steam is admitted to the intermediate-ahead chest, throttled to suitable pressure, and passes through the same paths as under full-speed conditions.

*Low Cruising Speeds.*—The reciprocating engines are used in connection with the turbines at these speeds. Steam is expanded through the reciprocating unit from which it is ex-

U.S.S. PARKER.

DIAGRAM ARRANGEMENT OF MACHINERY IN ENGINE ROOM.

A.- MAIN STEAM FROM BOILERS
B.- MAIN STEAM CROSS CONNECTION.
C.- STEAM TO CRUISING ENGINE.
D.- EXHAUST FROM CRUISING ENGINE.
E.- STEAM TO AHEAD TURBINE.
F.- STEAM TO ASTERN TURBINE.




1. LUBRICATING OIL PUMP.
2. TURBO GENERATOR.
3. OIL COOLER CIRCULATING PUMP.
4. OIL COOLER.
5. MAIN CIRCULATING PUMP AND ENGINE.
6. MAIN CONDENSER.
7. AHEAD TURBINE.
8. ASTERN TURBINE.
9. MAIN FEED PUMPS.
10. RECIPROCATING CRUISING ENGINE.
11. FRICTION CLUTCH.
12. REHEATER.
13. AUX. CONDENSER, COMB. AIR & CIRC. PUMP.
14. FEED AND FILTER TANK.
15. TORSIONMETERS.
16. FIRE AND BILGE PUMPS.
17. MAIN AIR PUMPS.
18. EVAPORATOR PLANT.
19. AIR COMPRESSOR.

FIG. 1.

hausted into the reheater, thence to the intermediate-ahead chest, through the turbines and into the condenser. The heating agent in the reheaters is the exhaust steam from the reciprocating-engine cylinder jackets.

For astern motion at all speeds the astern turbines alone are used, and for maneuvering the turbine units only are in use.

An interesting feature of the design of machinery in this class of boat built at the Cramp's yard is the "change valve." By means of this valve the supply of steam to either of or both the main and intermediate-ahead chests in the interests of economy or regulation of speed, may be regulated.

## SHAFTING AND BEARINGS.

There are two lines of shafting, as shown in Fig. 1, each in seven sections, namely, the crank shaft for the reciprocating engines; the intermediate shaft connecting the reciprocating and turbine units; the rotor shaft, within the turbine casing, supported by the thrust bearings; two sections of line shafting, supported by spring bearings; the stern-tube shaft, supported by the stern-tube bearings, and one propeller shaft, supported by the strut bearings. All shafting is of Class "A" steel forging.

*Data.*

| | |
|---|---|
| Crank shaft, length, feet and inches | 5–07⅛ |
| diameter, outside, inches | 05½ |
| axial hole, inches | 02 |
| Intermediate shaft, length, feet and inches | 9–05$\frac{9}{32}$ |
| diameter, outside, inches | 05¼ |
| Rotor shaft, length, feet and inches | 19–00¾ |
| diameter, outside, at bearings, inches | 11¼ |
| axial hole, inches | 07 |
| Line shaft, length, forward section, feet and inches | 10–04½ |
| after section, feet and inches | 21–11¼ |
| diameter, outside, inches | 08½ |
| axial hole, inches | 05¼ |
| Stern-tube shaft, length, feet and inches | 25–07¼ |
| diameter, outside, inches | 08⅝ |
| axial hole, inches | 05¼ |

| | |
|---|---|
| Propeller shaft, length, feet and inches | 32–01 |
| diameter, outside, inches | 08⅝ |
| axial hole, inches | 05¼ |
| Bearings, rotor shaft, number of | 2 |
| length, forward, inches | 16½ |
| after, inches | 16½ |
| intermediate shaft, number of | 1 |
| length, inches | 10 |
| cruising engine, number of | 3 |
| length, forward, inches | 07½ |
| middle, inches | 09 3/32 |
| after, inches | 09⅛ |
| Collars on shaft, number of | 9 |
| thickness of, inch | 00¾ |
| diameter outside, inches | 12½ |
| inside, inches | 08 11/16 |
| bearing surface, ahead, square inches | 493.8 |
| astern, square inches | 555.5 |
| Thrust shoes, number of, ahead | 8 |
| astern | 9 |
| effective thrust surface, ahead, square inches | 463.0 |
| effective thrust surface, astern, square inches | 520.9 |
| Line-shaft bearings, number of | 2 |
| length, inches | 16 |
| Stern-tube bearings, length, forward, inches | 27 |
| after, inches | 43 |
| Propeller-strut bearings, length, forward, inches | 27 |
| after, inches | 43 |

## *Shaft Calibration Data.**

| | |
|---|---|
| Length of shaft, calibrated, starboard, feet and inches | 10–00½ |
| Length of shaft, calibrated, port, feet and inches | 10–00½ |
| Length of shaft which torsion meter records, starboard, feet and inches | 10–00½ |
| Length of shaft, which torsion meter records, port, feet and inches | 10–00½ |
| Foot pounds for one inch distance between marks on cards, starboard | 29,473. |
| port | 29,258. |
| Horsepower constant,† starboard | 5.6117 |
| port | 5.5707 |

* For U. S. S. *Parker* only.

† S.H.P. equals: Constant multiplied by distance in inches between marks on card multiplied by the revolutions per minute.

*Main Bearings.*—There is one main bearing at each end of each turbine, the pedestal of which is cast with the turbine casing. The bottom brass is arranged to roll out without lifting the shaft after the weight of it has been taken up with the lifting gear, and the top brass to come off with the cap.

*Thrust Bearings.*—All turbines are provided, at the forward end, with thrust bearings, carried in the cylinder casting. It is halved, and so designed that the upper half will take the steam thrust and the lower half the propeller thrust. The collars are of brass, semicircular in shape, let into bushings and caulked into place, fitted into corresponding collars on the shaft.

*Spring Bearings.*—There are four spring bearings for the line shafting, of the self-oiling type, secured to foundations built upon the frame of the ship.

*Turning Gear.*—Each shaft is fitted with hand-operated turning gear, carried by brackets on the turbine casing. It consists of a worm wheel upon the shaft, meshing with a worm operated by a ratchet wrench arranged so that the gear may be readily connected or thrown out.

*Lifting Gear.*—An efficient hand-operated gear is provided for the main turbines.

## PROPELLERS.

There are two three-bladed propellers. The starboard propeller is right-hand and the port is left. The hubs and blades are of manganese bronze, cast solid. The blades are true screw machined to pitch and polished.

### *Propeller Data.*

| | |
|---|---|
| Diameter, feet and inches | 7–08½ |
| Pitch, feet and inches | 6–08 |
| Ratio of pitch to diameter | .8649 |
| Area, projected, square feet | 28.208 |
| helicoidal, square feet | 31.50 |
| disc, square feet | 47.77 |
| Ratio projected to disc | .605 |
| helicoidal to disc | .675 |

## MAIN CONDENSING APPARATUS.

*Main Condenser.*—There are two main condensers of elliptic-triangular section, located outboard of the low-pressure turbines. They are of the bent-tube type, with the tubes expanded into the tube sheets. The principal dimensions are as follows:

| | |
|---|---|
| Maximum width, inside, feet and inches | 4–04½ |
| depth, inside, feet and inches | 8–00¾ |
| Length between tube sheets, feet and inches | 12–06 |
| Thickness of tube sheets, inches | 01½ |
| Tubes, number | 2,934 |
| diameter, inch | 00⅝ |
| thickness, B.W.G. | 18 |
| Cooling surface, square feet | 6,000.9 |
| Exhaust nozzle, area through, square feet | 19.4 |
| Diameter of air-pump dry suction, inches | 08½ |
| wet suction, inches | 07½ |
| circulating-water inlet and outlet, inches | 18½ |

## MAIN AIR PUMPS.*

There are two Blake vertical, twin, beam-bucket air pumps, located in the auxiliary room just abaft the forward bulkhead, and immediately inboard the two shafts.

### *Data of One Main Air Pump.*

| | |
|---|---|
| Diameter of steam cylinder, inches | 14 |
| water cylinders, inches | 28 |
| Stroke of pistons, inches | 18 |
| Diameter of piston rods, inches | 02⅜ |
| suction nozzle, inches | 11 |
| discharge nozzle, inches | 09 |
| steam pipe, inches | 02½ |
| exhaust pipe, inches | 03 |

## MAIN CIRCULATING PUMPS.*

For each condenser there is a single-inlet centrifugal circulating pump, driven by a single-cylinder inclosed reciprocating engine. These pumps are located on the extreme outboard sides of the engine room, just forward of the condensers.

*See Table 2.

# TABLE 2.

## PUMPS AND PUMP CONNECTIONS.

## U.S.S. PARKER.

| NUMBER | NAME | CYLINDERS. STEAM. NO. | STEAM. DIAM INS. | WATER. NO. | WATER. DIAM INS. | STROKE (INCHES) | TYPE. | PIPES. SUCTION. INS. | SUCTION. FROM | DISCHARGE. INS. | DISCHARGE. TO | LOCATION. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 2 | MAIN AIR | 1 | 14 | 2 | 28 | 18 | TWIN VERTICAL, BEAM, BUCKET. | 11 | CONDENSER THROUGH WATER SEAL. | 9 | FILTER CHAMBER OF FEED TANK. | Aux. Room. |
| 2 | MAIN CIRCULATING. | 1 | 8 | | | | ENGINE DRIVEN CENTRIFUGAL. | 18½<br>7 | SEA.<br>BILGE. | 18½<br>3 | MAIN CONDENSER.<br>OIL COOLER (STBD. ONLY) | Eng. Room. |
| 2 | MAIN FEED. | 1 | 15 | 1 | 10 | 16 | VERTICAL SIMPLEX PISTON. | 5½<br>5½ | FEED SUCTION PIPE AND RESERVE FEED TANK.<br>WATER SEAL CROSS CONNECTION | 4 | MAIN FEED DISCHARGE. | do do |
| 2 | AUXILIARY FEED. | 1 | 15 | 1 | 10 | 16 | do do do | 5½<br>1½ | FEED SUCTION PIPE AND RESERVE FEED TANK.<br>HOSE CONNECTION. | 4<br>1½ | AUXILIARY FEED DISCH.<br>HOSE CONNECTION. | ONE EACH FIRE ROOM. |
| 2 | FIRE AND BILGE. | 1 | 7 | 1 | 7 | 12 | do do do | 4<br>4<br>1½ | SEA.<br>DRAINAGE.<br>HOSE CONNECTION. | 2½<br>3½<br>2<br>1½<br>1½ | FIRE MAIN.<br>OVERBOARD.<br>DISTILLER CIRC.<br>AIR COMPRESSOR CIRC.<br>HOSE CONNECTION.<br>FLUSHING SYSTEM THROUGH FIRE MAIN. | AUX. ROOM |
| 1 | AUXILIARY CONDENSER. | 1 | 6 | 2 | 8 | 7 | HORIZONTAL SIMPLEX COMBINED AIR AND CIRCULATING. | 4<br>4 | AUXILIARY CONDENSER.<br>SEA. | 3<br>4 | FEED TANK.<br>AUXILIARY CONDENSER. | do do |
| 1 | EVAPORATOR FEED | 1 | 4½ | 1 | 6 | 6 | VERTICAL SIMPLEX PISTON. | 2<br>1½ | SEA<br>DISTILLER CIRC. WATER DISCH. | 1½ | EVAPORATORS. | do do |
| 1 | DISTILLER FRESH WATER | 1 | 3½ | 1 | 4 | 4 | do do do | 1½ | DISTILLER RESERVOIR TANK. | 1⅛<br>1⅛<br>1⅛ | RESERVE FEED TANK<br>SHIPS TANKS.<br>COFFERDAM. | do do |
| 1 | LUBRICATING OIL. | 1 | 4½ | 1 | 6 | 6 | do do do | 3 | LUBRICATING OIL TANK. | 2½ | BEARINGS. | Eng. Room |
| 1 | OIL COOLER CIRCULATING | 1 | 7 | 1 | 7 | 12 | do do do | 3 | SEA. | 2½ | OIL COOLER. | do do |
| 4 | FUEL OIL SERVICE | 2 | 6 | 2 | 3½ | 8 | do DUPLEX do | 2½<br>2½ | OIL STORAGE TANKS.<br>BOOSTER DISCHARGE. | 2 | BURNERS. | TWO EACH FIRE ROOM |
| 2 | FUEL OIL BOOSTER | 1 | 4½ | 1 | 6 | 6 | do SIMPLEX do | 3<br>4<br>1¼ | OIL STORAGE TANKS.<br>DECK HOSE CONNECTION<br>OIL TANKS (WATER DRAIN) | 2½<br>2½<br>2½<br>2½ | SERVICE PUMP SUPPLY<br>STORAGE TANKS.<br>DECK HOSE CONNECTION<br>SEA (OIL TANK WATER). | ONE EACH FIRE ROOM |

*Data of One Main Circulating Pump.*

| | |
|---|---|
| Diameter of impeller, inches | 33 |
| Width of impeller at tip of blades, inches | |
| Diameter of suction nozzle, inches | 19½ |
| discharge nozzle, inches | 18½ |
| steam cylinder, inches | 08 |
| piston rod, inches | 01¼ |
| Stroke of piston, inches | 06 |
| Diameter of steam pipe, inches | 01½ |
| exhaust pipe, inches | 02 |

*Feed and Filter Tank.*—There is located in the auxiliary room on the port side abaft the forward bulkhead a feed and filter tank of 874 gallons capacity. In the top of the tank is located the filter tank of 166 gallons capacity, so arranged that the water is admitted to and delivered from the top of the chambers, thus keeping them filled with water and the filtering material submerged. A three-quarter-inch connection is provided for the admission of lime-water solution.

ENGINE ROOM AUXILIARIES.

*Main Feed Pumps.**—Located in the port forward corner of the engine room are two Blake vertical simplex, piston, double-acting pumps, 15 inches × 10 inches × 16 inches. These pumps are provided with suctions from the main feed tanks and discharge to the boilers through the feed heaters or through by-passes.

*Auxiliary Condenser, Air and Circulating Pumps.**—The auxiliary condenser, of 256 square feet cooling surface, is in the port after corner of the auxiliary room. Immediately below, and connected to the condenser, is a Blake 6 inches × 8 inches × 8 inches × 7 inches horizontal, simplex, combined air and circulating pump. The condenser is connected to all the machinery through the auxiliary exhaust line.

*Forced-Lubrication System.*—The main bearings, thrust bearings and main circulating-pump engines are provided with forced lubrication.

*See Table 2.

FIG. 2.

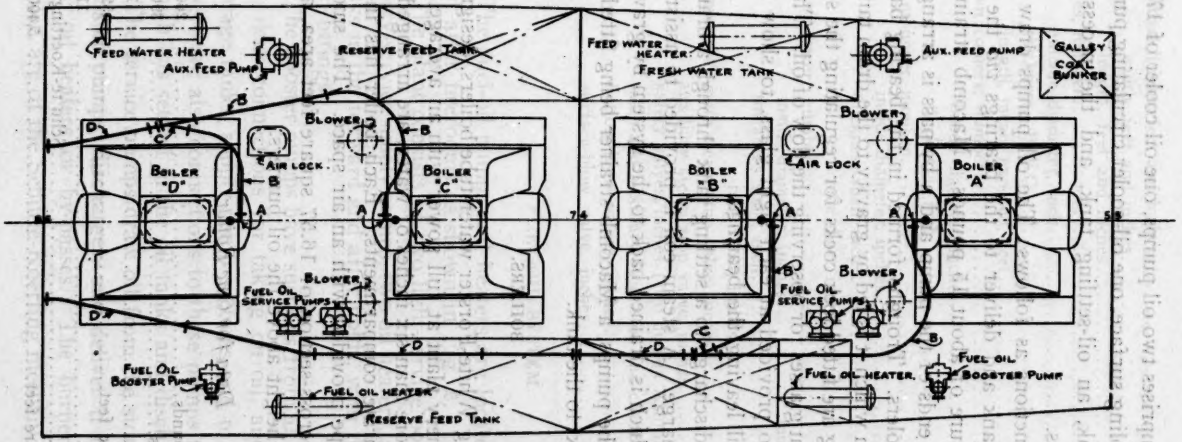



DIAGRAM ARRANGEMENT OF MACHINERY IN FIRE ROOM
U.S.S. PARKER

A – 7" MAIN BOILER STOP VALVE.
B – 7" MAIN STEAM FROM BOILER.
C – 9½" × 7" × 7" EXPANSION JOINT.
D – 9½" MAIN STEAM LINES.
E – 9½" STUFFING BOX.

The system comprises two oil pumps, one oil cooler of 175.2 square feet of cooling surface, one oil-cooler circulating pump, one oil-drain tank, an oil-settling tank, and the necessary piping and fittings.

The system functions as follows: The oil pumps draw oil from the drain tank and deliver to the bearings *via* the oil coolers at a pressure of about 15 pounds. Macomb strainers are fitted at both ends of the pump and a by-pass is arranged around the oil coolers. Troughs formed in the bearing bases catch the oil, from which it is led by gravity to the drain tanks.

At each bearing are fitted lock cocks for regulating the supply of oil, and sight glasses for observing the flow of oil. Thermometers are also provided at each sight glass, to show the temperature of oil leaving the bearings.

The oil pumps discharge to a settling tank through a branch of the main discharge. A steam coil is provided to assist in settling the oil, which is drained back to the system by gravity or drawn off by the pumps, a Macomb strainer being fitted in the connection next to the tank.

## BOILERS.

Four oil-burning White-Forster water-tube boilers, designed to run the machinery plant at full power with an average air pressure of not more than six inches of water, are arranged in pairs in two separate compartments. Each boiler has independent smoke pipe provided with an air space. The smoke pipes are of oval cross-section of 16.57 square feet area, and are 30.05 feet in height above the oil burners.

*Data for One Boiler.*

| | |
|---|---|
| Working pressure, pounds | 260 |
| Test pressure, pounds | 390 |
| Number of furnaces | 1 |
| Furnace volume, cubic feet | 452 |
| Oil burners, number | 11 |
| type | Schu'te-Koerting. |
| Heating surface, square feet | 5,400 |

| | |
|---|---|
| External height, feet and inches | 12–07 7/16 |
| width, feet and inches | 14–07 3/8 |
| length, feet and inches | 13–02 1/16 |
| Tubes, number | 3,256 |
| outside diameter, inch | 01 |
| thickness, B.W.G. | 13 |
| length, mean, exposed, feet | 6.34 |
| Down comers, number | 2 |
| inside diameter, inches | 08 1/8 |
| thickness, inch | 00 3/8 |
| Drum, upper, number | 1 |
| inside diameter, inches | 48 |
| thickness shell, inch | 00 9/16 |
| tube sheet, inches | 01 9/16 |
| lower, number | 2 |
| inside diameter, inches | long, 30; short 19 1/2 |
| thickness shell, inch | 00 9/16 |
| tube sheet, inches | 01 9/16 |
| Diameter of main steam stop valve, inches | 07 |
| auxiliary steam stop valve, inches | 04 |
| safety valves, inches (two duplex Ashton) | 04 |
| main and auxiliary feed, stop and check valves, ins. | 03 |
| bottom-blow valves, inches | 01 1/2 |
| surface-blow valve, inches | 01 1/4 |

## FUEL-OIL SYSTEM.

The fuel-oil pumps and heaters are located in the forward and after firerooms as shown in Fig. 2.

The plant consists of two light-service booster pumps, four heavy-service supply pumps, two oil heaters, and the oil-storage tanks in the forward and after holds, with the necessary pipes and fittings.

The booster pumps have suctions from all storage tanks and the deck connections for taking on oil, and discharge to all storage tanks, supply-pump suctions and to deck connections. There are also connections to these pumps to draw the water which has settled in the oil tanks and discharge it overboard.

The service pumps draw oil from the storage tanks and the booster pumps' discharges, and deliver it to the burners *via* the oil heaters or by-passes. The burners, eleven to each boiler, are of the Schutte-Koerting mechanical atomizing type.

Strainers are fitted in the pump suctions and discharges, and the supply pipes to burners are provided with cocks controlled from both the fireroom and deck.

For raising steam when no air pressure is available, there is provided in each fireroom a small hand pump designed to deliver oil to two burners at about 200 pounds pressure.

### FIREROOM AUXILIARIES.

*Forced-Draft Blowers.*—There are two blowers for each fireroom. These fans are of the Keith single-inlet, horizontal type, mounted on vertical shafts in the base of the fireroom ventilators, from which the air supply is taken. The blowers in the after firerooms are provided with ducts drawing air from the engine room. These ducts are provided with dampers operated from the fireroom floor. Each fan is driven by a 24-inch Terry turbine located immediately below the fan. The fan data are as follows:

| | |
|---|---|
| Fan, diameter, inches | 35⅛ |
| width, inches | 11 $\frac{13}{16}$ |
| number of blades | 24 |
| Diameter of inlet, inches | 32½ |

*Feed-Water Heater.*—In each fireroom there is a feed-water heater of the dead-end type, installed on the discharge side of both the main and auxiliary feed pumps. The heating agent is the exhaust steam, a back pressure being kept in the auxiliary exhaust line for this purpose. The principal dimensions of each heater are as follows:

| | |
|---|---|
| Diameter shell, inside, inches | 17 |
| Length between tube sheets, feet and inches | 6–05⅛ |
| Tubes, number | 238 |
| diameter, inch | 00⅝ |
| thickness, inch | 0.049 |
| Heating surface, square feet | 250.0 |
| Diameter of feed inlet and outlet, inches | 05½ |
| auxiliary exhaust nozzle, inches | 07 |
| drain, inches | 03 |

*Auxiliary Feed Pumps.**—In each fireroom there is a Blake, 15-inch × 10-inch × 16-inch auxiliary feed pump. They are of the vertical, simplex, piston type, and so arranged as to feed any of the boilers, either through the feed heaters or direct.

### MAIN STEAM PIPING.

There are two main steam lines. The port line is connected to the two after boilers only, and the starboard line to the two forward boilers. The main steam line is 9½ inches in diameter, with 7-inch branch pipes to the boilers. The general arrangement of the main steam piping is shown in Figs. 1 and 2.

### AUXILIARY STEAM PIPING.

There is a 4-inch stop valve on each boiler for supplying the auxiliary steam line, which leads throughout the machinery space with connections to the various auxiliaries, and a branch leading forward to the deck machinery, forward heating system and galleys.

### AUXILIARY EXHAUST PIPING.

An auxiliary exhaust line is led throughout the machinery space and elsewhere, as required, for the various auxiliaries. Connections are provided for direct exhaust into either main or auxiliary condensers, either feed-water heater, or into the atmosphere through the after escape pipe. There are also connections for admitting exhaust steam into the 11th stage of each turbine, when so desired.

### MAIN AND AUXILIARY FEED SYSTEM.

From the feed tank, a pipe is led to the stop valves on the main feed pumps in the engine room and on auxiliary feed pumps in the fireroom.

The main feed pumps will discharge through a 5½-inch main leading to the boiler compartments. This pipe branches in the

*See Table 2.

after fireroom into a 4-inch pipe for the boilers in that compartment, and into another 4-inch pipe leading to the forward fireroom for the boilers there. From these branches 3-inch pipes are led to the individual boilers. All feed pumps are arranged to feed any boiler *via* the main feed line, and through the feed heater or by-passes. In addition, each auxiliary feed pump has direct connection to the boilers in its own fireroom.

### INTERIOR COMMUNICATION.

The customary engine and fireroom telegraphs, gongs, telephones, voice tubes, etc., are fitted for transmitting orders and signaling to the machinery compartments and other parts of the vessel.

### EVAPORATING AND DISTILLING APPARATUS.

The evaporators are located in the auxiliary room, as shown in Fig. 1, with the distillers located in the after engine-room hatch at a high elevation.

There are two evaporators and two distillers, with their accessories, arranged to operate in single effect. The plant has a combined capacity of 3,000 gallons of water per twenty-four hours.

*Data for One Evaporator.*

| | |
|---|---|
| Type | Vertical, Reilly, multicoil. |
| Diameter, inside, inches | 32 |
| Height, over all, feet and inches | 6–06 |
| Number of tube coils | 10 |
| Tubes in coils, diameter, inch | 01 |
| thickness, B.W.G. | 16 |
| Heating surface, square feet | 61.5 |
| Diameter of steam nozzle, inches | 01½ |
| vapor nozzle, inches | 02½ |
| feed valve, inch | 01 |
| blow valve, inches | 01¼ |

*Data for One Distiller.*

| | |
|---|---|
| Type | Modified Bu. S.E. |
| Diameter, inside, inches | 09 |
| Length between tube sheets, feet and inches | 2–07 |

| | |
|---|---|
| Tubes, number | 49 |
| diameter, inch | 00⅝ |
| thickness, B.W.G. | 18 |
| Cooling surface, square feet | 20.54 |
| Diameter of circulating water inlet and outlet, inches | 01½ |
| Diameter of vapor inlet, inches | 02 |
| drain, inch | 01 |

### MACHINE TOOLS.

For performing minor repairs one small Hendey-Norton motor-driven engine lathe, of 12 inches swing and 6-foot bed, is installed, together with all the necessary tools and attachments.

### ELECTRIC PLANT.

The dynamos are located in the forward starboard corner of the engine room. The installation consists of two horizontal, compound-wound, direct-current, 5-kilowatt, General Electric generators, each driven by a Curtis steam turbine. Each generator will deliver 40 ampères at 125 volts, when making 5,000 revolutions per minute.

### TORSIONMETERS.

An improved type of Gary-Cummings torsionmeter is provided for each line of shafting, for ascertaining the shaft horsepower developed at the various speeds.

### TRIALS.

The contract trials were as follows:

(*a*) A progressive trial over the measured-mile course, for standardizing the screws.

(*b*) A full-speed trial of four hours duration in the open sea, in deep water, at the highest speed attainable, the average speed to be not less than 29.5 knots, with the average air pressure in the firerooms not exceeding 6 inches of water and the steam pressure at the high-pressure turbine not to exceed 240 pounds above the atmosphere.

| NUMBER OF RUN | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 | 26 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| TIME ON COURSE, MINUTES | 5 | 4 | 5 | 3 | 4 | 5 | 3 | 2 | 3 | 2 | 2 | 2 | 2 | 2 | 2 | 2 | 2 | 2 | 1 | 2 | 1 | 2 | 1 | 1 | 1 | 1 |
| SECONDS | 15.9 | 50.2 | 25.5 | 41.1 | 9.3 | 35.4 | 23.9 | 53.5 | 25.6 | 32.0 | 54.3 | 32.7 | 19.1 | 41.1 | 20.6 | 19.5 | 6.1 | 18.7 | 58.1 | 5.2 | 56.6 | 1.6 | 57.3 | 29.9 | 57.3 | 56.4 |
| REVOLUTIONS TO GO 1 KNOT | | | | | | | | | | | | | | | | | | | | | | | | | | |
| STARBOARD SHAFT | 1038.8 | 947.2 | 1056.8 | 939.5 | 1063.0 | 918.1 | 1080.3 | 916.0 | 1095.6 | 952.7 | 1092.8 | 954.5 | 980.7 | 1121.1 | 980.8 | 1172.5 | 1053.0 | 1152.6 | 1142.2 | 1220.0 | 1155.8 | 1233.5 | 1194.3 | 1223.4 | 1207.4 | 1210.0 |
| PORT SHAFT | 1040.4 | 958.7 | 1064.8 | 940.8 | 1063.9 | 918.5 | 1087.9 | 921.3 | 1092.3 | 948.0 | 1095.3 | 953.4 | 981.1 | 1127.0 | 985.6 | 1172.4 | 1050.3 | 1158.6 | 1141.4 | 1228.0 | 1153.6 | 1235.9 | 1186.1 | 1230.2 | 1199.5 | 1218.3 |
| AVERAGE | 1039.6 | 953.0 | 1060.7 | 940.0 | 1063.5 | 918.3 | 1088.1 | 919.7 | 1094.0 | 950.4 | 1095.9 | 954.0 | 980.9 | 1124.1 | 982.3 | 1172.5 | 1051.7 | 1155.6 | 1141.8 | 1224.0 | 1154.7 | 1234.7 | 1190.2 | 1226.8 | 1203.5 | 1212.7 |
| REVOLUTIONS PER MINUTE | | | | | | | | | | | | | | | | | | | | | | | | | | |
| STARBOARD SHAFT | 197.26 | 198.90 | 195.18 | 255.08 | 256.41 | 256.01 | 320.89 | 317.49 | 319.70 | 376.10 | 376.54 | 375.03 | 422.99 | 417.37 | 418.43 | 504.28 | 499.86 | 495.86 | 580.91 | 584.69 | 583.66 | 608.84 | 610.87 | 611.83 | 618.08 | 613.36 |
| PORT SHAFT | 197.56 | 194.15 | 196.73 | 255.33 | 255.62 | 255.90 | 319.86 | 318.63 | 318.72 | 374.26 | 377.54 | 374.62 | 423.16 | 419.59 | 419.73 | 504.24 | 498.58 | 501.43 | 580.10 | 587.11 | 584.02 | 609.72 | 606.66 | 618.25 | 614.03 | 616.13 |
| AVERAGE | 197.47 | 196.03 | 195.95 | 255.20 | 255.52 | 255.96 | 320.22 | 318.06 | 319.21 | 375.18 | 377.04 | 374.83 | 423.08 | 418.98 | 419.08 | 504.26 | 499.22 | 500.14 | 580.51 | 585.90 | 583.84 | 609.13 | 608.77 | 613.54 | 616.04 | 614.72 |
| SPEED IN KNOTS | 11.596 | 12.405 | 11.067 | 16.282 | 16.440 | 16.713 | 17.756 | 20.749 | 17.510 | 23.684 | 20.684 | 23.876 | 25.881 | 22.344 | 25.605 | 25.806 | 25.649 | 25.955 | 30.483 | 28.754 | 30.303 | 29.605 | 30.691 | 30.205 | 30.691 | 30.408 |
| S.H.P. STARBOARD SHAFT | 332 | 341 | 329 | 691 | 687 | 704 | 1349 | 1364 | 1381 | 2301 | 2345 | 2315 | 3442 | 3420 | 3405 | 6197 | 6031 | 5861 | 8112 | 8268 | 8257 | 8913 | 9016 | 8893 | 9122 | 9082 |
| PORT SHAFT | 352 | 324 | 329 | 725 | 740 | 725 | 1443 | 1402 | 1420 | 2473 | 2482 | 2400 | 3748 | 3623 | 3648 | 5899 | 5888 | 6006 | 8357 | 8275 | 8361 | 9035 | 9159 | 9220 | 9238 | 9301 |
| TOTAL | 684 | 665 | 658 | 1416 | 1427 | 1429 | 2792 | 2766 | 2801 | 4774 | 4827 | 4715 | 7190 | 7043 | 7053 | 12096 | 11919 | 11857 | 16469 | 16543 | 16618 | 17948 | 18175 | 18113 | 18360 | 18383 |

TABLE-3. — STANDARDIZATION TRIAL DATA — U.S.S. PARKER — COURSE OFF DELAWARE BREAKWATER ON NOVEMBER 18, 1913.

TABLE 3.

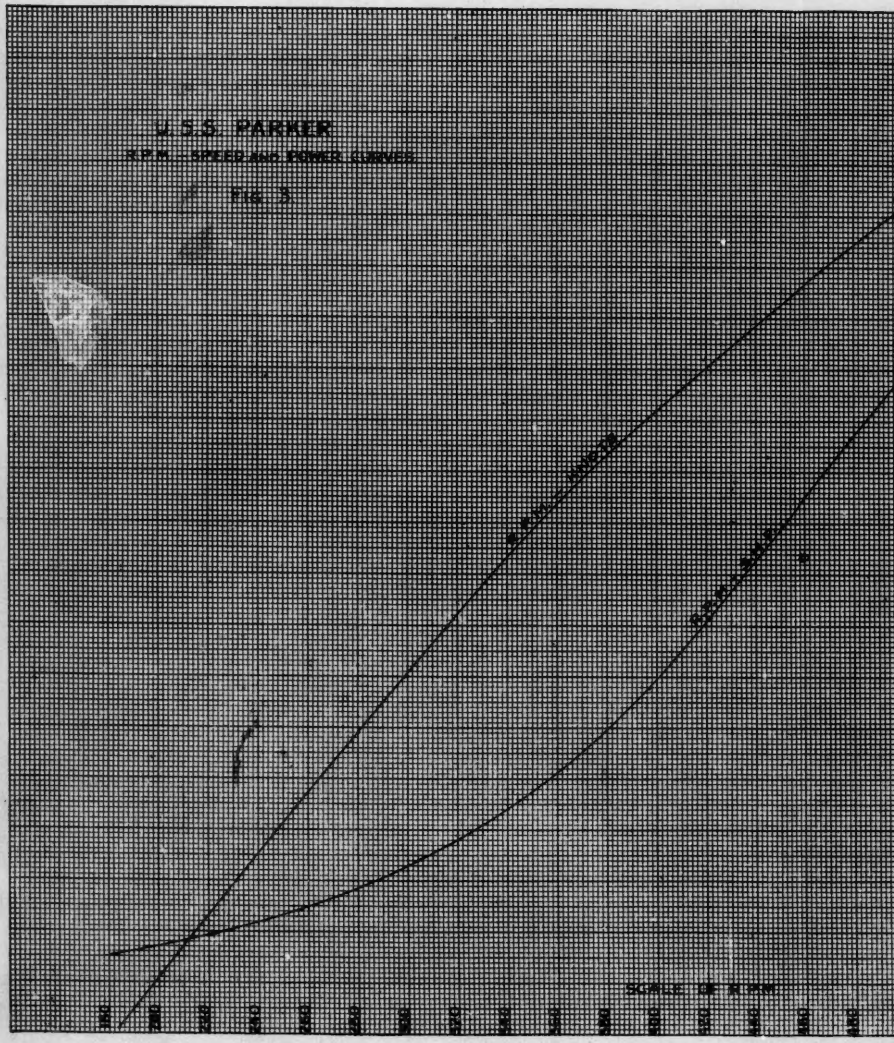



FIG. 3.

KEUFFEL & ESSER CO., NEW YORK, NO. 354 S.

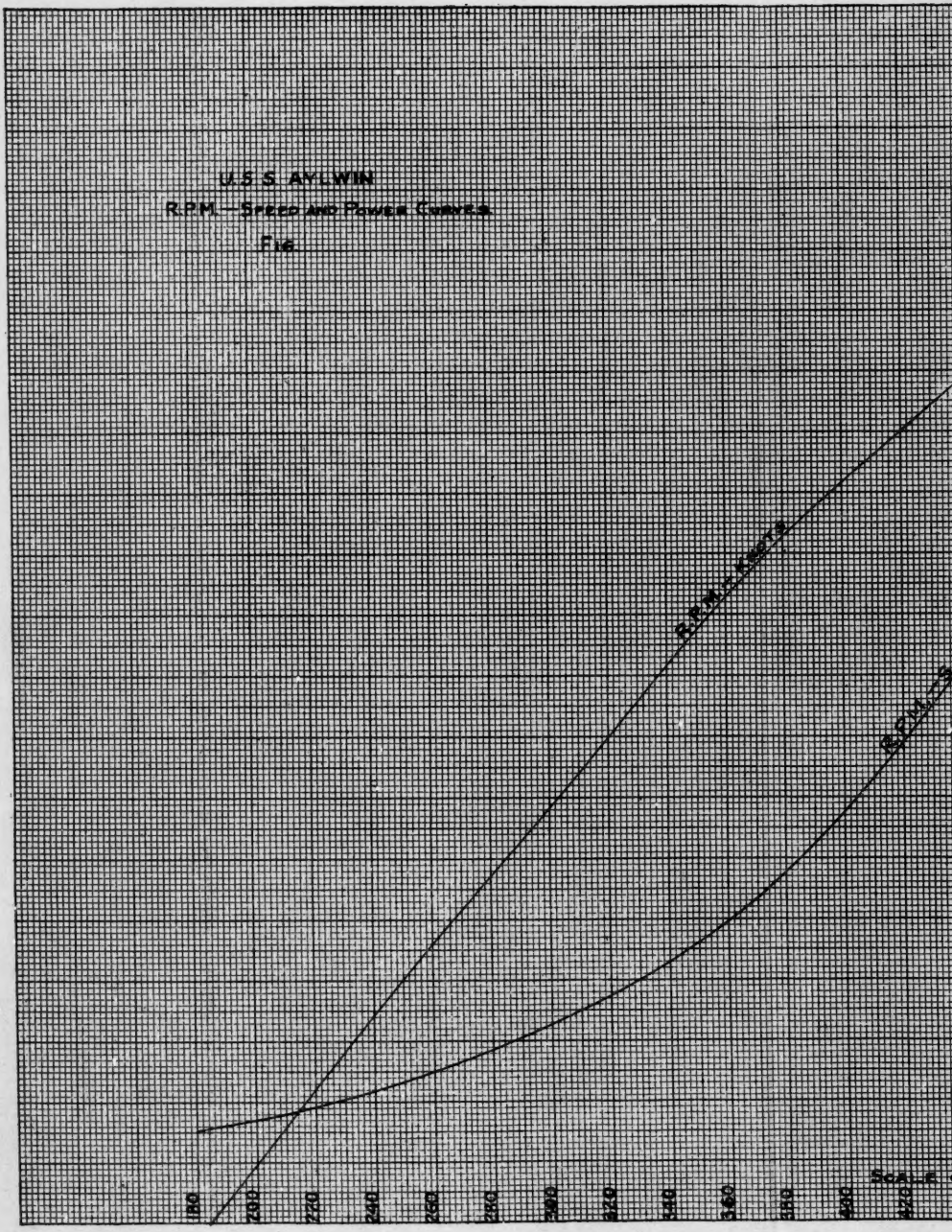

U.S.S. AYLWIN
R.P.M. – SPEED AND POWER CURVES.
FIG.
H.P. AT PROPELLERS
H.P. AT ENGINES
180
200
220
240
260
280
300
320
340
360
380
400
420
SCALE

FIG. 4.

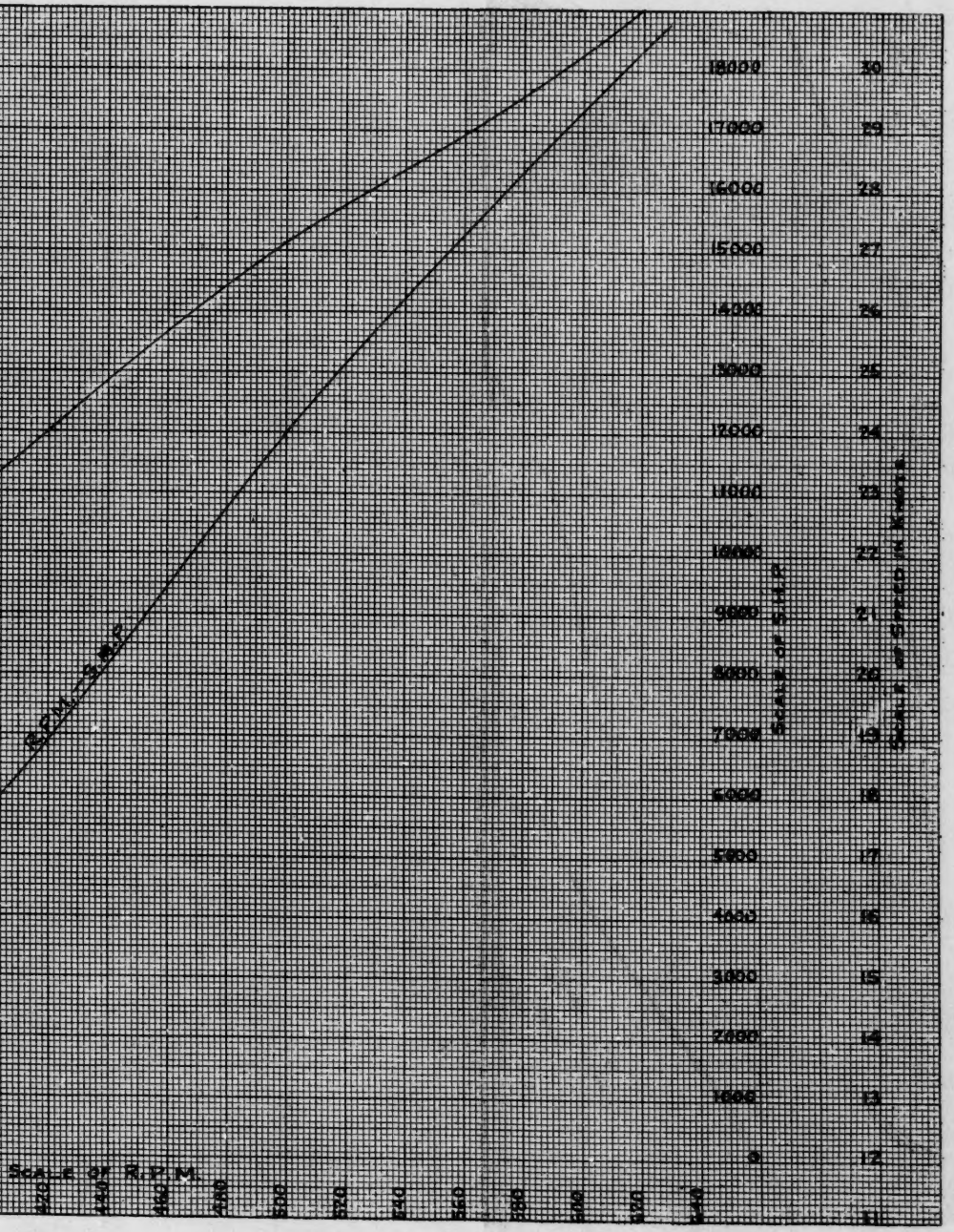

18000
17000
16000
15000
14000
13000
12000
11000
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
0
Scale of H.P.
Scale of Torque
Scale of R.P.M.

FIG. 4.

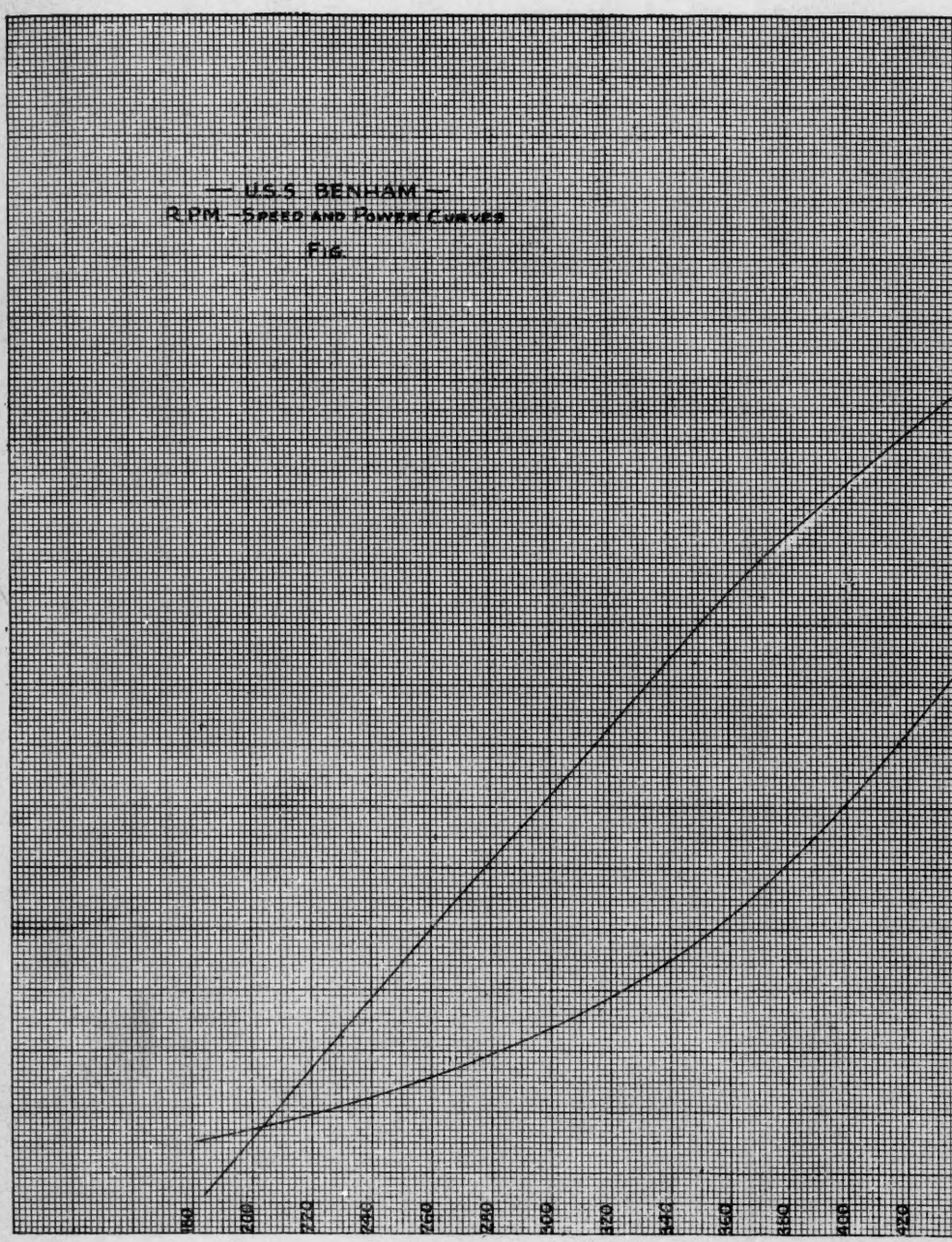

— U.S.S. BENHAM —
R.P.M.—Speed and Power Curves
Fig.
180
200
220
240
260
280
300
320
340
360
380
400
420

Fig. 5.

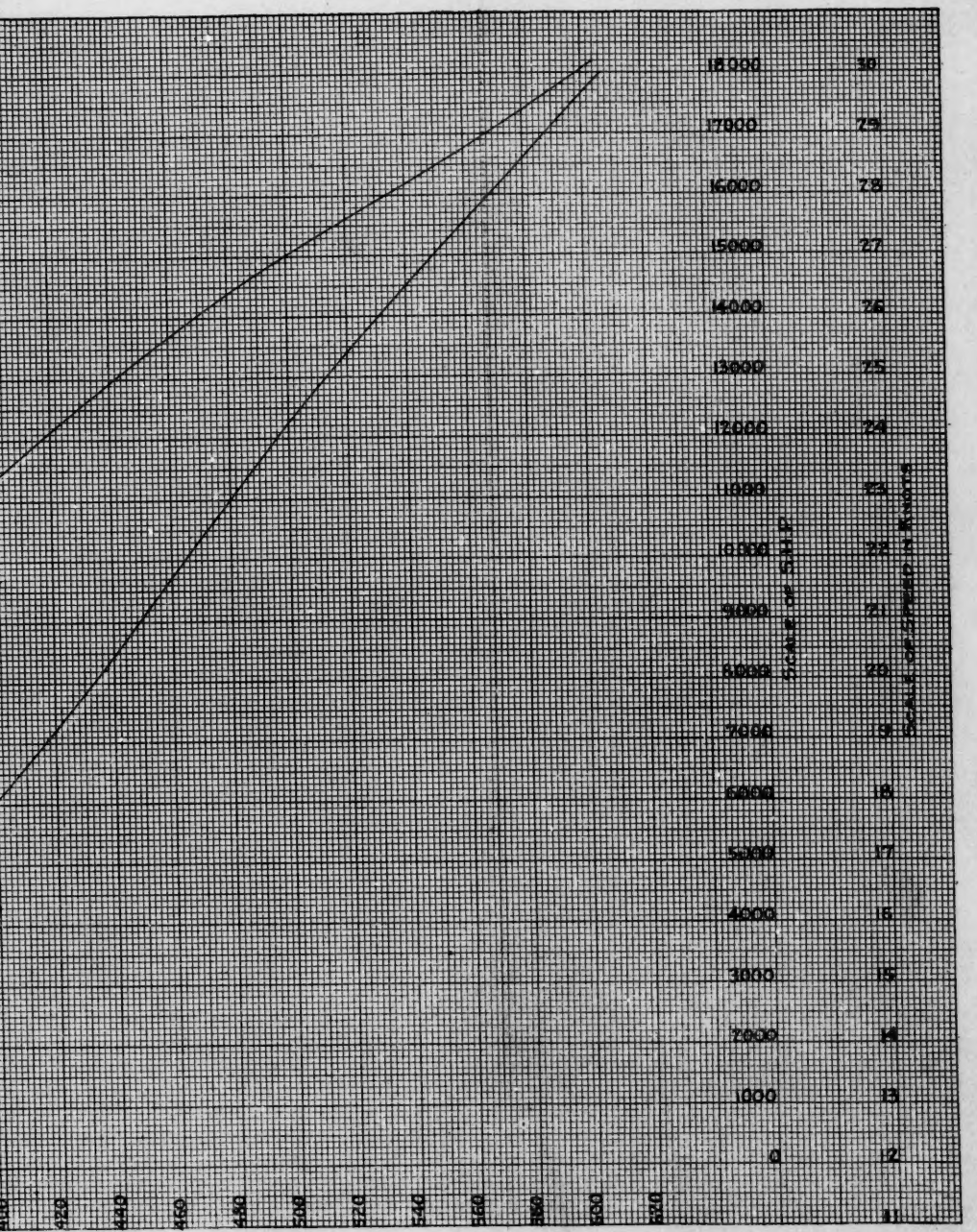

18000
17000
16000
15000
14000
13000
12000
11000
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
0
30
29
28
27
26
25
24
23
22
21
20
19
18
17
16
15
14
13
12
11
Scale of H.P.
Scale of Revolutions per minute
420
440
460
480
500
520
540
560
580
600
620

FIG. 5.

(*c*) A fuel-oil and water-consumption trial of four hours duration in the open sea at an average uniform speed of 24 knots as nearly as possible.

(*d*) A fuel-oil and water-consumption trial of four hours duration in the open sea at an average speed of 15½ knots as nearly as possible. This trial to be made as nearly as possible under service conditions, with the cruising engines connected up and in use.

(*e*) An endurance trial of twenty hours duration in the open sea at an average speed of 15½ knots as nearly as possible, under service cruising conditions, with cruising engines connected up and in use.

(*f*) A fuel-oil and water-consumption trial of four hours duration in the open sea with the cruising engines connected up and in use, at an average speed of 12 knots as nearly as possible.

The standardization trials were run over the measured mile off the Delaware Breakwater. On each vessel twenty-six runs were made over the course, the data from which were used in plotting the curves shown in Figures 3, 4 and 5. An example of this data, for the *Parker,* is shown in Table 3.

It was determined from the official speed and revolution curves that the following revolutions per minute were required to obtain the contract speeds on the endurance trials:

| Speed, in knots. | *U.S.S. Parker.* R.p.m. | *U.S.S. Aylwin.* R.p.m. | *U.S.S. Benham.* R.p.m. |
|---|---|---|---|
| 12 | 198.6 | 201.4 | 191.1 |
| 15.5 | 256.0 | 258.7 | 250.8 |
| 24 | 418.8 | 419.8 | 417.7 |
| 29.5 | 581.3 | 577.8 | 584.0 |

The endurance trials were run in the open sea off the Delaware Capes. The dates and the data obtained on these trials are given in Tables 4, 5, 6 and 7.

## TABLE 4.—DATA ON FOUR-HOUR FULL-POWER TRIAL.

| | *Aylwin.* | *Benham.* | *Parker.* |
|---|---|---|---|
| Date of trial | Oct. 16, 1913 | Dec. 17, 1913 | Nov. 19, 1913 |
| Speed, in knots | 29.595 | 29.594 | 29.547 |
| Draught, mean on trial, feet and inches | 9-3½ | 9-4½ | 9-4⅛ |
| Displacement, corresponding, tons | 1,020 | 1,035 | 1,035 |
| Engines in operation | Turbines. | Turbines. | Turbines. |
| Number boilers in use | 4 | 4 | 4 |
| oil burners in use, average per hour | 41.1 | 41.9 | 42.3 |
| Air pressure in firerooms, inches of water | 6.6 | 6.86 | 7.12 |
| Heating surface used, square feet | 21,600 | 21,600 | 21,600 |
| Pressures (average) : | | | |
| main steam at boilers, gage | 248.3 | 249.4 | 248.7 |
| full ahead steam chest, gage | 215.5 | 215 | 213 |
| Vacuum, condenser, top, mercurial, inches | ... | 27.49 | 26.62 |
| bottom, gage, inches | 27.925 | 28.79 | 28.24 |
| Barometer | 29.95 | 30.04 | 30.08 |
| Injection, temperature, degrees F | 63.9 | 56.9 | 59.4 |
| Main feed pressure, pounds gage | 337.2 | 345.9 | 300.9 |
| temperature, degrees F | 195.1 | 183.3 | 193.3 |
| Revolutions per minute of propellers, average | 580.48 | 585.32 | 583.08 |
| starboard | 581.24 | 584.98 | 582.89 |
| port | 579.71 | 585.65 | 583.27 |
| Collective S.H.P. from curve | 16,350 | 16,850 | 16 500 |
| computed* | 16,286 | 16,610 | 16,680 |
| Slip of propellers, in per cent. of own speed, mean | 20.52 | 23.28 | 22.99 |
| starboard | 20.62 | 23.23 | 23.01 |
| port | 20.41 | 23.33 | 22.96 |
| Water, pounds per hour † | 260,323 | 283,633.25 | 247,486.75 |
| per S.H.P | 15.984 | 17.075 | 14.837 |
| Fuel oil, pounds, per hour | 21,161.28 | 20,786 | 20,822.5 |
| per knot | 715.028 | 703.44 | 713.1 |
| per S.H.P | 1.299 | 1.251 | 1 248 |
| B.t u., per pound | 19,114 | 19,500 | 19,446 |
| temperature to burners, degrees F | 124.9 | 123.4 | 125 |

* Used in computing oil and water consumptions. †On the *Benham*, water measured represents that used for all purposes, and on the *Aylwin* and *Parker*, water used in the main turbines alone.

## TABLE 5.—FOUR-HOUR 24-KNOT TRIAL.

| | *Aylwin.* | *Benham.* | *Parker.* |
|---|---|---|---|
| Date of trial | Oct. 16, 1913 | Dec. 17, 1913 | Nov. 19, 1913 |
| Speed in knots | 24.086 | 24.023 | 24.103 |
| Draught, mean on trial, feet and inches | 9-3⅛ | 9-5⅛ | 9-4⅛ |
| Displacement, corresponding, tons | 1,025.5 | 1,045 | 1,040 |
| Engines in operation | Turbines. | Turbines. | Turbines. |
| Number boilers in use | 3 | 4 | 4 |
| oil burners in use, average per hour | 18.9 | 19.75 | 20.6 |
| Air pressure in firerooms, inches of water | 3.8 | 3.62 | 3.31 |
| Heating surface used, square feet | 16,200 | 21,600 | 21,600 |
| Pressures (average). | | | |
| main steam at boilers, gage | 245.9 | 245.5 | 244.1 |
| medium ahead steam chest, gage | 202 | 215 | 213.7 |
| Vacuum condenser, top, mercurial, inches | ... | 28.54 | 28.18 |
| bottom, gage, inches | 28.67 | 28.32 | 29.00 |
| Barometer | 29.92 | 29.91 | 30.08 |
| Injection, temperature, degrees F | 62.9 | 53.3 | 55.3 |
| Main feed, pressure, pounds gage | 327 | 354.4 | 290.1 |
| temperature, degrees F | 199.2 | 189.6 | 209 3 |
| Revolutions per minute of propellers, average | 421.8 | 418.12 | 421.07 |
| starboard | 421.9 | 417.87 | 421.38 |
| port | 421.7 | 418.37 | 420.07 |
| Collective S.H.P. from curve | 7,025 | 7,090 | 7,150 |
| computed* | 7,212 | 7,290 | 7,287 |
| Slip of propellers, in per cent. of own speed, mean | 10.99 | 12.67 | 13.0 |
| starboard | 11.01 | 12.62 | 13.07 |
| port | 10.97 | 12.77 | 12.93 |
| Water, pounds per hour † | 125,818.5 | 138,491.0 | 120,425 |
| per S.H.P | 17.446 | 18.997 | 16.256 |
| Fuel oil, pounds per hour | 9,783 86 | 9,429.1 | 9,560.5 |
| per knot | 406.205 | 392.5 | 396.7 |
| per S.H.P | 1.356 | 1.293 | 1 312 |
| B.t.u., per pound | 19,114 | 19,500 | 19,550 |
| temperature to burners, degrees F | 131.8 | 152.9 | 152.2 |

* Used in computing oil and water consumptions. †On the *Benham*, water measured represents that used for all purposes, and on the *Aylwin* and *Parker*, the water used in the main turbines alone.

## Table 6.—Four-Hour 15.5-Knot Trial.

| | *Aylwin.* | *Benham.* | *Parker.* |
|---|---|---|---|
| Date of trial | Dec. 9, 1913 | Dec. 18, 1913 | Nov. 20, 1913 |
| Speed, in knots | 15.61 | 15.544 | 15.578 |
| Draught, mean on trial, feet and inches | 9-4 | 9-6¼ | 9-5⅛ |
| Displacement corresponding, tons | 1,025 | 1,045 | 1,038.4 |
| Engines in operation | Combination of turbines and reciprocating. | | |
| Number boilers in use | 2 | 2 | 2 |
| oil burners in use, average per hour | 6.125 | 5 | 6 |
| Air pressure in firerooms, inches of water | 1.36 | 1.8 | 1.48 |
| Heating surface used, square feet | 10,800 | 10,800 | 10,800 |
| Pressures (average): | | | |
| main steam at boilers, gage | 246.7 | 240.9 | 246.8 |
| H.P. valve chest, gage | 210.9 | 225.6 | 230.7 |
| Vacuum, condenser, top, mercurial, inches | 29.15 | 28.36 | 28.50 |
| bottom, gage, inches | 29.52 | 28.86 | 29.02 |
| Barometer | 30.26 | 29.91 | 30.09 |
| Injection, temperature, degrees F. | 43.9 | 46.1 | 51.8 |
| Main feed, pressure, pounds gage | 315.6 | 331.3 | 341.9 |
| temperature, degrees F. | 197.4 | 210.3 | 214.4 |
| Revolutions per minute of propellers, average | 260.52 | 251.63 | 257.24 |
| starboard | 260.67 | 251.17 | 257.12 |
| port | 260.38 | 252.09 | 257.36 |
| Collective S.H.P. from curve | 1,500 | 1,430 | 1,450 |
| computed* | 1,510 | 1,399 | 1,472 |
| Indicated horsepower, reciprocating engines | 855 | 780 | 841 |
| Slip of propellers, in per cent. of own speed, mean | 6.60 | 6.11 | 7.96 |
| starboard | 6.65 | 5.94 | 7.92 |
| port | 6.54 | 6.28 | 8.00 |
| Water, pounds per hour† | 2,774.92 | 32,281.25 | 26,758.25 |
| per S.H.P. | 18.371 | 23.075 | 18.178 |
| Fuel oil, pounds per hour | 2,501.96 | 2,190.8 | 2,286 |
| per knot | 160.279 | 140.94 | 146.7 |
| per S.H.P. | 1.657 | 1.566 | 1.553 |
| B.t.u. per pound | ... | 19,500 | 19,550 |
| temperature to burners, degrees F. | 173.4 | 158.1 | 156.4 |

*Used in computing oil and water consumptions. †On the *Benham*, water measured represents that used for all purposes, and on the *Aylwin* and *Parker*, the water used in the main turbines alone.

## Table 7.—Four-Hour 12-Knot Trial.

| | *Aylwin.* | *Benham.* | *Parker.* |
|---|---|---|---|
| Date of trial | Oct. 15, 1913 | Dec. 16, 1913 | Nov. 18, 1913 |
| Speed, in knots | 11.985 | 12.010 | 11.990 |
| Draught, mean on trial, feet and inches | 9-5⅛ | 9-5⅞ | 9-4¾ |
| Displacement corresponding, tons | 1,055 | 1,040.8 | 1,035.7 |
| Engines in operation | Combination of turbines and reciprocating. | | |
| Number boilers in use | 2 | 2 | 2 |
| oil burners in use, average per hour | 3.8 | 2.63 | 4 |
| Air pressure in firerooms, inches of water | 1.6 | 1.8 | 1.87 |
| Heating surface used, square feet | 10,800 | 10,800 | 10,800 |
| Pressures (average): | | | |
| main steam at boilers, gage | 242.7 | 224.9 | 232.5 |
| H.P. valve chest, gage | 139.6 | 124.9 | 146.6 |
| Vacuum, condenser, top, mercurial, inches | ... | 28.78 | 28.10 |
| bottom, gage, inches | 28.1 | 29.39 | 28.56 |
| Barometer | 30.01 | 30.04 | 39.10 |
| Injection temperature, degrees F. | 61.1 | 46 | 53.6 |
| Main feed pressure, pounds gage | 340 | 333.8 | 343.8 |
| temperature, degrees F. | 212.8 | 199.5 | 221.6 |
| Revolutions per minute of propellers, average | 200.1 | 192.76 | 198.48 |
| starboard | 200.3 | 192.64 | 198.64 |
| port | 199.8 | 192.87 | 198.32 |
| Collective S.H.P. from curve | 700 | 630 | 690 |
| computed* | 626 | 645 | 625 |
| Indicated horsepower, reciprocating engine | 434 | 378 | 453 |
| Slip of propellers, in per cent. of own speed, mean | 7.07 | 4.59 | 8.19 |
| starboard | 7.17 | 4.53 | 8.26 |
| port | 6.97 | 4.64 | 8.11 |
| Water, pounds, per hour† | 20,188 | 19,716.5 | 14,464.25 |
| per S.H.P. | 32.249 | 29.828 | 23.143 |
| Fuel oil, pounds per hour | 1,643.97 | 1,358.84 | 1,617.5 |
| per knot | 137.858 | 112.29 | 134.9 |
| per S.H.P. | 2.626 | 2.056 | 2.588 |
| B.t.u., per pound | 19,114 | 19,500 | 19,550 |
| temperature to burners, degrees F. | 158.7 | 181.2 | 174.2 |

*Used in computing oil and water consumptions. †On the *Benham*, water measured represents that used for all purposes, and on the *Aylwin* and *Parker*, the water used in the main turbines alone.

# SEARCHLIGHTS ON BOARD MODERN WAR VESSELS.*

Translated from the Italian of CAPTAIN E. PINELLI by P. SMILES, ESQ., R.N., with discussion by LIEUT. LOGAN CRESAP, MEMBER.

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Up to now the number and position of the searchlights on board our men-of-war have been based on no sound principles, whether in regard to the special conditions under which they are employed, the position of the look-out stations, or the arcs of training of the anti-torpedo-boat guns.

The problem has always been put forward in such a way that we have been forced to make use of the existing state of affairs on board in order to determine the best positions for the searchlights. When we bear this restriction in mind, it is obvious that we are practically deprived of all choice in the matter, as we can do no more than make the best of the possible situations afforded us by the funnels and masts, which alone offer, especially in modern ships, clear and elevated positions. On this account almost all navies have been driven to employ platforms built on the sides of the funnels, a solution which is by no means satisfactory.

For obvious reasons the projectors ought to be stowed in protected positions during the day, but it is to be feared that as time goes on the searchlights, which are extremely delicate instruments, will deteriorate, and this state of things will be aggravated still more if, as we have every reason to believe, searchlights increase in size and number on board future ships.

Besides, even when the lights are struck down behind armor in the day time, their mountings, which are indispensable to them, remain always exposed to damage from the enemy's fire, and from the blast of their own big guns, so that we run the

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* "Journal Royal United Service Institute."

risk of keeping the projectors themselves safe, but of having nowhere to mount them when the day action is over.

When choosing positions for the lights it would be of inestimable advantage if they were not tied down to any one point in the ship, so that they could be grouped together as much as possible at the best angle of visibility, as we must bear in mind that it has for some time been admitted that the most favorable position for a searchlight, with regard to the guns with which it is working, is that which maintains as nearly as possible an angle of 30 degrees between the line of fire, and the beam of light which illuminates the target under fire. It is obvious that these conditions cannot be carried out in their entirety on board a ship, so that we shall have to limit ourselves to keeping the searchlight positions as far as possible from the batteries of anti-torpedo-boat guns.

On board ship as well as on shore it pays to keep the duties of the different lights absolutely distinct, and when we consider the conditions under which these work we see the necessity of having two categories of lights—"Searching Lights," whose duty it will be to sweep the horizon slowly, uniformly, and with fixed elevation, and "Gun Lights," to illuminate a target which has already been picked up.

The problem of the best means of disposing searchlights on board a man-of-war is closley bound up with the way in which we propose to employ them in the case of a torpedo attack. This is a problem already long discussed, and on which the following opinions are generally held:

(1) The best way of avoiding torpedo attacks is to keep the ship darkened.

(2) All ships ought to have a perfect look-out service, and in this connection it is difficult to understand why, granted the great importance of this service in the case of a sudden attack by torpedo craft, it is still considered that any seaman without special training is competent to perform this all-important duty. It would rather seem necessary to institute a special branch of "Look-outs," chosen each year from the recruits gifted with the best eyesight, and well trained in picking up torpedo craft.

These "Look-outs" would keep watch in pairs from convenient stations divided into two groups and well protected from the weather so as to assure to them the most favorable possible conditions of visibility. The principal station would be in charge of a lieutenant who would be able to switch on all the lights on his own authority, and who would warn by a special signal the anti-torpedo-boat guns best placed for opening fire at once. This is certainly a big responsibility, but it is justified by the supreme importance of not losing even an instant in the most critical moments following the discovery of the enemy's torpedo craft proceeding to the attack.

When the hostile destroyer has been sighted it will be our aim to dazzle it by keeping it in the beam of one of our lights, and for this purpose a projector which *can be mechanically controlled from a distance* is practically indispensable.

(3) It is probable that the first boat to be picked up will only be making a feint attack, and that the decisive one may come from a totally different direction. The "Look-outs" would, therefore, have to be specially trained on this account never to let themselves be distracted from the most important task assigned to them, and to increase their vigilance in their own special sector.

(4) When the hostile craft has been finally picked up by a light, the commanders of groups of anti-torpedo-boat guns on the engaged side will open fire at once, without waiting for authority. These minutes are too precious to the safety of the ship for any other course to be taken.

(5) One of the "Gun Lights" will at once take in hand, so to speak, the enemy's craft upon which fire has already been opened, while the "Searching Light" will take up again its former duty of sweeping the horizon. It is obvious that when, proceeding as we have said, all the "Gun Lights" are employed in illuminating torpedo craft which are still dangerous, the "Searching Lights" will have to take over the duty of "Gun Lights," since there will no longer be anyone to relieve them when they have picked up a new destroyer.

(6) If a ship finds that she has been discovered for certain

by the enemy's torpedo craft, the captain with one simple and conventional signal will order all the "Searching Lights" to sweep, and in such a case they would do this together, their speed being regulated according to the length of the arc assigned to each. Each sector would be swept at a fixed angle of elevation, varying according to the height of the projector above the water line, so as to ensure that any hostile torpedo craft must be picked up, and will remain as long as possible under the fire of the small and medium-caliber guns before getting within practical torpedo range. The "Gun Lights" will in the meantime light up the torpedo boats as they are successively picked up in the sector assigned to them.

If we admit the advantages possessed by such a system of employing our searchlights, we see at once how in the case of a converging attack the number of projectors installed on board our modern ships is generally insufficient.

The height above water line which has been found to be in practice the most convenient for the lights is that of 8–12 meters for the "Searching Lights" and 30–35 meters for the "Gun Lights," and these heights are generally recognized as the best. Taking into consideration the great speed of modern torpedo craft, we should not assign an arc of a length greater than 45 degrees to any "Searching Light."

If we decide to employ our lights as has been explained shortly above, we shall need eight projectors for the "Searching Service," while in the case of the "Gun Lights" various necessities force us to limit their number. There should, however, never be less than four of them, that is to say, one for every arc of 90 degrees.

Granted the great importance of the modern warship of large displacement, we should remember that we ought to and must devote a greater space in it to searchlights, which, to all intents and purposes, take at night the place of the eyes of this colossus, whose Achilles' heel lies just in its very weakness against submarine attack.

A disposition which answers to the needs which we have

touched upon is that shown in the attached sketch. It comprises twelve lights, eight "Searching" and four "Gun" lights, all of 90 cms. and 150 ampères, and mechanically controlled from a distance. The "Searching Lights" are mounted on disappearing supports worked on the system of telescopic masts, which can be lowered by some simple mechanism into protected positions below the upper deck; the "Gun Lights" are installed on each mast. In the sketch are shown eight supports for the "Searching Lights," disposed in two groups, each of four, the one forward and the other aft. Each support can be locked into position at two different heights.

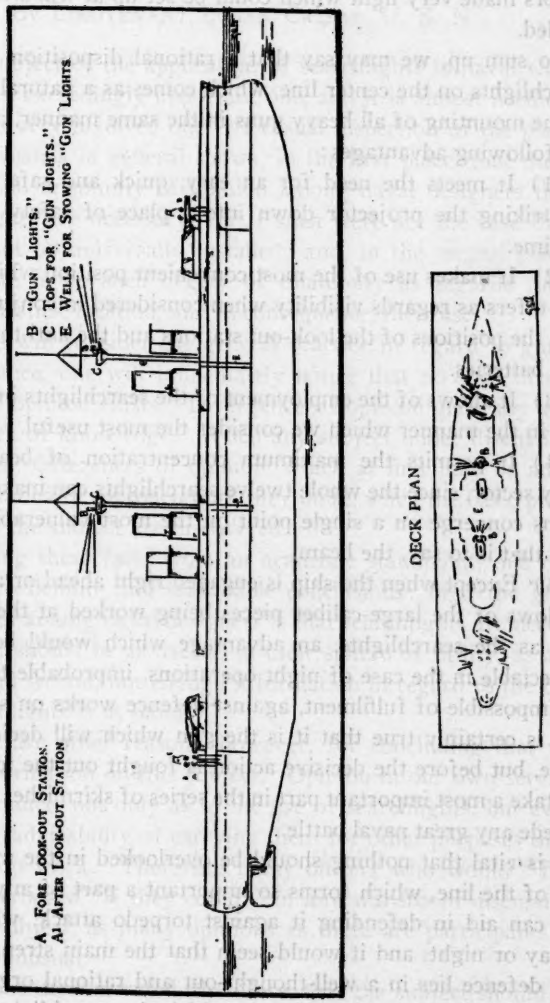
By this simple arrangement we obtain the important advantage of giving to each light separately, and to all of them together, a large field of illumination. So, for example, it is possible to direct the four beams of the lights of one group on one beam, by securing in the lower position the two supports on the side it is wished to illuminate, and locking the other two in the higher one; or one can in the same way arrange that all the lights in the foremost group should bear right aft. So, too, can the four lights of one group be trained parallel to one another on the same target, and in certain cases this might constitute an advantage of some importance.

Besides, all the eight "Searching Lights" would be able to light up a wide arc of the horizon by keeping their beams motionless, which might prove most useful when the ship is at anchor and well protected by nets or other obstructions.

Naturally each support would have to be fitted with a removable platform and all the other needful accessories. If a ship had the big guns on her forecastle and quarterdeck mounted in triple turrets, the supports could be placed so as to rise up between the guns when they are trained fore and aft, and to these they might be stayed.

The "Gun Lights" are in the sketch disposed in the tops, and lifts fitted for the purpose, and working up and down the masts, would serve as a means for striking the projectors down behind armor and for remounting them again at night.

In addition to these twelve fixed lights it would be of advan-


SIDE ELEVATION OF ITALIAN SUPER-DREADNOUGHT
A Fore Look-out Station.
A1 After Look-out Station
B "Gun Lights."
C Tops for "Gun Lights."
E Wells for Stowing "Gun Lights"
DECK PLAN

tage in very long ships to keep on board also two special projectors made very light which could be set up at will wherever needed.

To sum up, we may say that a rational disposition of the searchlights on the center line, which comes as a natural result of the mounting of all heavy guns in the same manner, affords the following advantages:

(1) It meets the need for an easy, quick and safe means of striking the projector down into a place of safety in the daytime.

(2) It makes use of the most convenient position which the ship offers as regards visibility when considered in conjunction with the positions of the look-out stations and the anti-torpedo-boat batteries.

(3) It allows of the employment of the searchlights in every case in the manner which we consider the most useful.

(4) It permits the maximum concentration of beams in every sector, since the whole twelve searchlights can make their beams converge on a single point on the most vulnerable sectors, that is to say, the beam.

(5) Except when the ship is engaged right ahead or astern, it allows of the large-caliber pieces being worked at the same time as the searchlights, an advantage which would be very appreciable in the case of night operations, improbable though not impossible of fulfilment, against defence works on shore.

It is certainly true that it is the gun which will decide the battle, but before the decisive action is fought out the torpedo will take a most important part in the series of skirmishes which precede any great naval battle.

It is vital that nothing should be overlooked in the modern ship of the line, which forms so important a part of any fleet, that can aid in defending it against torpedo attack, whether by day or night, and it would seem that the main strength of such defence lies in a well-thought-out and rational organization of the "look-out" service, in which the searchlight plays such an outstanding part.

## DISCUSSION OF CAPTAIN PINELLI'S ARTICLE.

By LIEUTENANT LOGAN CRESAP, U. S. N.

The subject of the application of searchlights to naval warfare is an exceedingly important one, and it is almost needless to add, a very involved one. A casual inspection of the plans of war vessels in general shows, in the first place, that there is practical unanimity of opinion among naval designers that searchlights are necessary, for if such were not the case they would not be universally installed; and, in the second place, that there is no such degree of unanimity as regards their characteristics, location, and in consequence, their use.

If one will consider the general practice in regard to guns, for instance, one will immediately notice that no such divergence of opinion exists. Divergence of opinion generally signifies lack of knowledge. When any subject under discussion is studied, differences of opinion, which at first may be antipodal, gradually approach until they unite, when the basic principles of the subject are discovered.

Viewing these facts from an academic standpoint, one can form two definite and reasonable conclusions: first, the general belief among naval designers is that searchlights are necessary and should be as useful, in their sphere of utility, as are guns; and second, not enough information in regard to the use of searchlights is at hand.

There are other reasons, however, for concluding that we need information on this subject. Opinion in our own service is at variance, not only as to the use of searchlights, but even as to the advisability of carrying them for other purposes than navigation work. There are many officers who would "fire them overboard" if they could. In any wardroom discussion of this subject, as many opinions as there are participants is nothing unusual.

It seems demonstrated, therefore, that the subject should be investigated much more carefully and scientifically than has been yet done, for no one, no matter what his present opinion on the subject may be, will admit that we know enough.

The article by Captain Pinelli is very much to the point, is most opportune, and it is to be hoped, will arouse discussion and further investigation.

Practically all naval men agree with Captain Pinelli that searchlights should be kept behind armor except when needed aloft, and his suggestions for facilitating the change of location necessary can easily be tried out. It should be noted, however, that the apparatus he suggests is just as much subject to destruction as would be the lights themselves were they to be left exposed. Convenient davit sockets, slinging arrangements on the lights themselves, and ordinary whips used with taut rope travelers might prove just as useful and much less costly and cumbersome.

The scheme of dividing the lights into two groups, "gun lights" and "searching lights," is also worthy of careful verification, and much information is already on hand on this point and needs only to be worked into shape for comparison. It is, however, a disastrous matter to have crossed beams, and Captain Pinelli's scheme seems open to that contingency. Careful experiment only will demonstrate how best to handle this subject of the division of lights. The extreme ends of the ship offer an excellent location for gun lights, provided they can be raised sufficiently. This location gives, particularly in the latest vessels, a good angle between the line of fire and axis of beam.

Another point on which our service pretty well agrees is that the best defense of a battleship is to be darkened, and there are many who carry it farther—herein lies the big bone of contention—and maintain that she should stay darkened. One thing is certain, when the night is dark, the destroyer has just as much trouble as the battleship. The position that a destroyer can be dazzled is absolutely untenable. A pair of tinted-glass spectacles completely destroys the dazzling of the sun itself, and they can be put on and off instantly. It is absolutely mandatory that we continue our joint maneuvers with battleships and destroyers and carefully settle the questions in regard to when to turn on lights, when to open fire,

how to handle the lights; in other words, to really determine what a battleship's night-defense shall be.

Captain Pinelli has assumed that the battleship must alone defend herself. This is by no means necessary. She may fight the enemy, but she need not light him; and it is on this point that really settles the question of how to defend a battleship at night, though it must also be remembered that she should be able to do both—probably the reason for Captain Pinelli's assumption.

The most striking of Captain Pinelli's suggestions is that all searchlights should be put on the center line. This, however, is not necessary in order to make all lights bear on either beam, which is probably a most desirable condition. It must not be lost sight of in working out a solution of the problem that it is from astern and especially ahead that the greatest danger from attack exists. The torpedo may not be fired from ahead or astern, but the attack is most easily begun from those directions.

One other point should be considered, we should not forget to work out a proper plan for our destroyers as well as our battleships.

In conclusion: there is room for improvement in our searchlights themselves, and it should be accomplished, together with the solution of the problem of how to use them to the best advantage.

# FRICTION CLUTCH AND OPERATING GEAR FOR CRUISING ENGINES AND TURBINES.

By J. F. Metten, Associate.

Note.—The development of a reliable clutch which can be instantly engaged or disengaged without requiring a stoppage of the connecting shafts or the bringing of the two shafts to the same speed, and which will transmit large powers with certainty and without the need of elaborate apparatus, has been a great desideratum, especially in naval engineering installations where cruising units are installed which are to be disconnected at high speeds. The clutch developed by Mr. Metten, Chief Engineer, Wm. Cramp & Sons Ship and Engine Building Co., appears to answer all the requirements in a simple, reliable and effective manner. This clutch has actually stood the test of service, and it works. It would appear that a clutch of this type would have a very useful field in connection with the machinery of submarines, where the jaw clutches now employed are far from giving satisfaction.—H. C. D.

The machinery specifications for Destroyers Nos. 43 to 50 called for an installation of turbines and cruising engines that could be operated either in combination or under turbines alone with engine standing idle, and further required means for connecting and disconnecting engines from turbine shafts without stopping the propelling machinery.

The installation of reciprocating engines for cruising purposes in combination with turbines designed for full speed had been previously carried out on several destroyers built abroad, and trials of these vessels proved that they were more economical at low speeds than the destroyers fitted with cruising turbines coupled to the main turbine spindles. The type of machinery specified for the above U. S. vessels, which were

the first of the 1,000-ton class, was expected to considerably increase their cruising radius, and these expectations were fully realized in the performance of seven vessels of this class that have completed their trials. Of these seven, four were fitted with various forms of jaw couplings between the cruising engines and main turbine, the requirements regarding disconnecting while under way being waived on account of there being no reliable clutch capable of meeting the original requirements that was available.

As contracts for four vessels of this class were awarded The William Cramp & Sons Ship and Engine Building Company, this firm developed and built a special clutch adapted for this particular service which was fitted on the four destroyers built by them and which has worked very satisfactorily. These clutches are hydraulically operated, the pressure medium being oil supplied by the regular oil pumps for turbine lubrication.

Fig. 1 shows a section through the clutch proper, attached to engine crank shaft with the extension shaft connecting to the end of main turbine spindle, including a special steady bearing for supporting the extension shaft. The driving member of the clutch consists of a circular frame, A, bolted to the crankshaft flange and carrying a fixed circular friction member, B, also a similar but movable friction member, C, which is permanently attached to the clutch frame through the flexible ring diaphragm, E, the space between this friction member and the frame of the clutch being made into a closed chamber by the head, F. This head is formed of a steel plate, and this chamber receives the operating oil which is supplied through the axial hole in the engine crank shaft. The clutch is shown in released condition, and it will be evident that the application of pressure to the chamber, G, will cause the extension of the diaphragm until the driven disc, D, is clamped at its periphery between the two friction surfaces. It will also be evident that the load carried by the friction surfaces will be equal to the total load on the movable member less the resistance of the diaphragm to extension, and that the friction due to this load is doubled, as there are two sliding surfaces involved in any

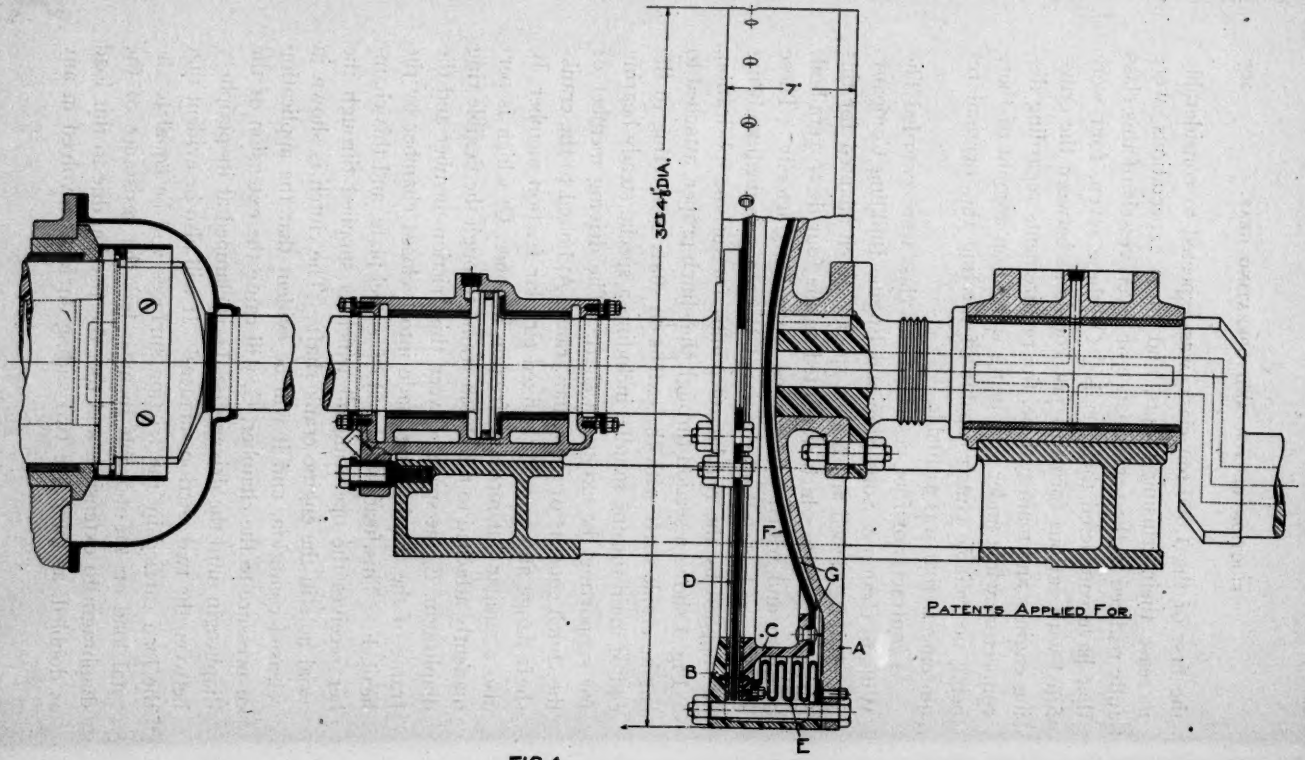



FIG. 1.

FIG. 1

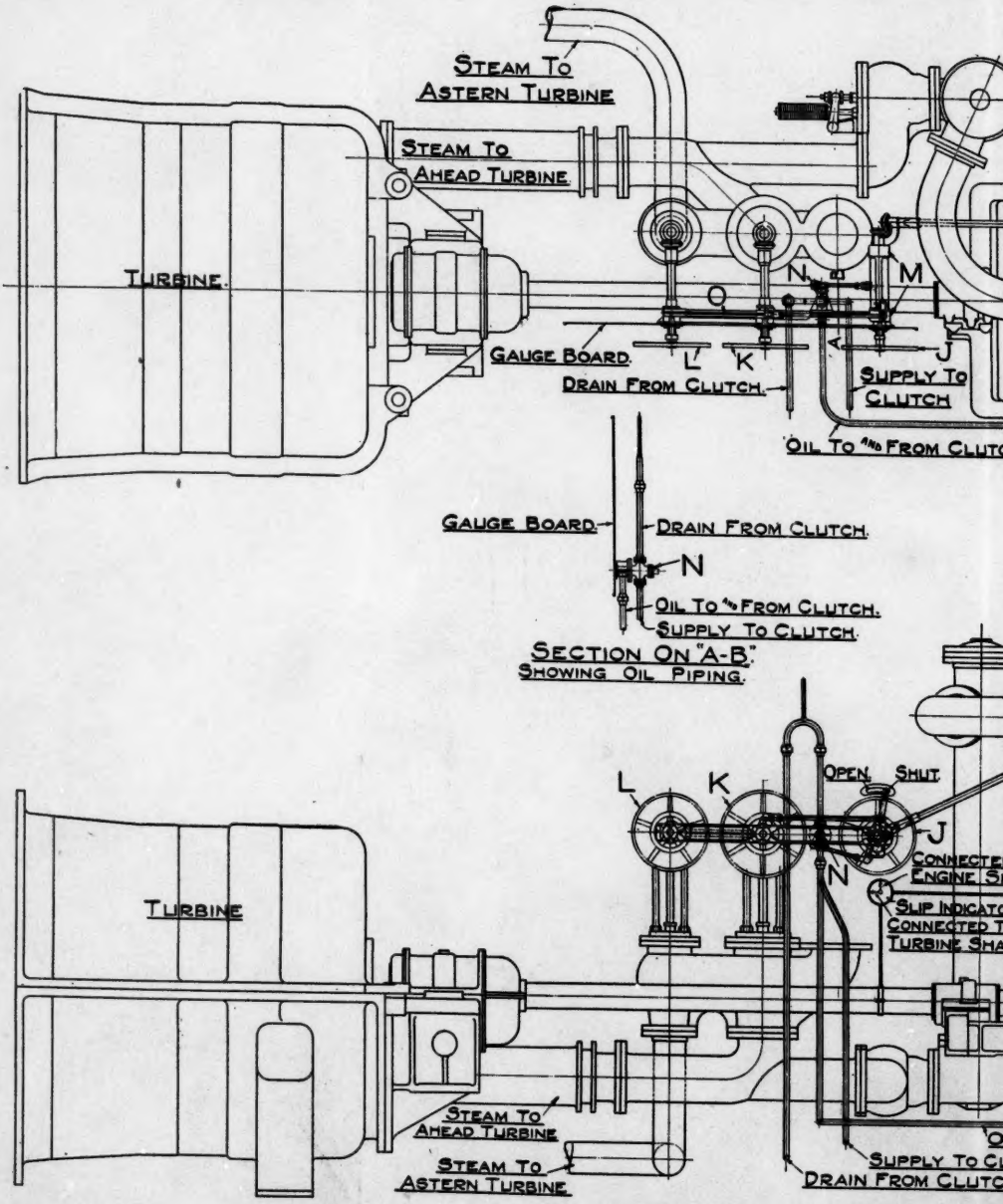
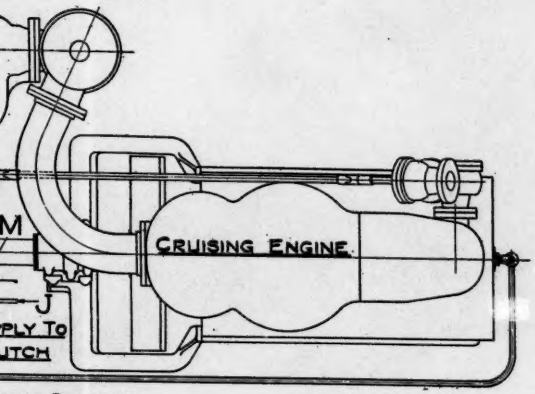
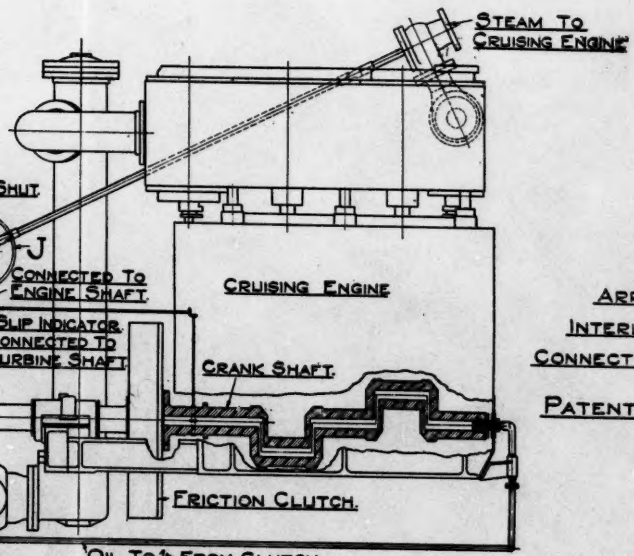

STEAM TO ASTERN TURBINE
STEAM TO AHEAD TURBINE
TURBINE
GAUGE BOARD
DRAIN FROM CLUTCH
SUPPLY TO CLUTCH
OIL TO AND FROM CLUTCH
N
M
O
L
K
J
GAUGE BOARD
DRAIN FROM CLUTCH
N
OIL TO & FROM CLUTCH
SUPPLY TO CLUTCH
SECTION ON "A-B"
SHOWING OIL PIPING
L
K
OPEN
SHUT
J
N
TURBINE
STEAM TO AHEAD TURBINE
STEAM TO ASTERN TURBINE

FIG. 2.




ARRANGEMENT OF
INTERLOCKING GEAR AND
CONNECTIONS TO FRICTION CLUTCH

PATENTS APPLIED FOR.

. 2.

movement of the driven member with relation to the driving member. The friction surfaces are lined with similar material to that used for automobile-brake linings which, in addition to having a high friction coefficient, is very durable.

A clutch of the size shown is capable of transmitting 1,500 B.H.P. (uniform torque) at 300 r.p.m. with a pressure of 35 pounds per square inch on the chamber. The clutch is operated by a three-way cock connecting the chamber to the pressure line while engaged and connecting same to a free outlet when released. The extension of the diaphragm must be such as to permit its elasticity being maintained in order that the contraction of the diaphragm will expel the oil from the chamber and give a quick release. The running clearance between disc D and the friction surfaces when idle is made about 3/32 inch on each side, which means that there is a difference in the plane of rotation of the disc amounting to 3/32 inch between the engaged and idling conditions. In order to insure the disc running central and free of the friction faces the steady bearing supporting intermediate shaft is fitted with a central collar thrust arranged so that the lubricating oil pressure in this bearing keeps the collar against the forward face and maintains the correct idle-running position of the disc. When, however, the clutch is engaged the collar leaves its face an amount equal to the longitudinal displacement of the disc's plane of rotation. The connection of the intermediate shaft to the turbine spindle is made by a sliding coupling in order to allow for the longitudinal expansion of the turbine from its anchorage at after end.

Fig. 2 shows the arrangement of operating gear for clutch, which is controlled entirely by the cruising-engine regulating-valve hand wheel J. It will be noted that the latter is connected to the engine regulating valve by an extension shaft, and is located beside the main ahead turbine regulating-valve wheel K and the astern turbine regulating-valve wheel L.

Mounted on the engine regulator spindle is an interrupted-screw device, M, which causes the outside sleeve to rotate to the left with the handwheel during the initial opening move-

ment and remains locked in this position through all points of the throttle opening up to the final closing position, when it rotates with the wheel to the right and is locked in that position until the valve is again opened. This sleeve operates the three-way cock, N, controlling the clutch and also operates with the same movement the locking bar, O, fitted to engage the main ahead and astern turbine regulators. When it is desired to go ahead under the cruising engine all that is required is to turn the cruising regulating-valve handwheel. The first movement applies the clutch before the valve is cracked off its seat, and the clutch can not be released except by returning the handwheel to its closed position. The interlocking bar, O, is free to move longitudinally only when both ahead and astern turbine valves are closed, and must be in the right-hand position before either of the turbine valves can be opened; opening either valve locks the bar in this position. It will be evident that the combination results in the following safeguards against improper handling:

1. Clutch can be engaged only by opening cruising-engine supply valve.

2. Clutch can not be released when this valve is open.

3. Release can be effected only by shutting off steam to the cruising engine.

4. Neither of the main turbine regulating valves can be opened while clutch is engaged.

5. The clutch can not be engaged nor steam turned on cruising engine when either of the turbine valves are open or partly open.

This interlocking and operating gear is extremely simple and enables the combination machinery being handled as quickly as the main turbines. As the operation of the clutch is a function of the cruising-engine regulating gear, from ahead under cruising engine in combination with turbine to astern under turbine alone involves only shutting the cruising-engine valve and opening the astern turbine valve. The cruising engine is fitted with a small by-pass and direct exhaust to keep

the engine turning over idly when under no load and for warming-up purposes.

It will be noted from the arrangement, Fig. 2, that the reciprocating engine is placed some distance from turbine, the object being to provide as much flexibility between engine and turbine journals as possible. This design of clutch provides additional advantages in this direction, as the driven disc is only ¼ inch in thickness and will permit of an appreciable lack of truth in shaft alignment without affecting running conditions. A slipping indicator is fitted near the operating gear, which consists of a dial with two pointers, one of which is driven by the engine shaft and one by turbine spindle.

While the clutches are designed to transmit the full load with approximately 40 pounds oil pressure, they are tested to over 100 pounds per square inch and can be subjected to an operating pressure of 75 pounds per square inch. There is no loss of oil, the diaphragm and other joints being well bolted. During preliminary trials on one vessel the friction surfaces of one clutch were purposely flooded with oil, which easily carried its full load by increasing the oil pressure to 60 pounds. By treating the friction rings with kerosene and applying a little powdered rosin the normal operating pressure could be returned to immediately. It is not necessary to have the engine and turbine running at the same speeds when engaging this type of clutch, as approximately five seconds elapse from the time operating cock is opened until the full load is applied to the friction surfaces, so the load is taken gradually. The displacement from released to applied conditions amounts to approximately one gallon of oil, and this amount must flow in and out of the chamber each time the clutch is applied or released. The time required for these functions can be regulated by increasing or decreasing the opening through which oil is supplied. Shop tests of the first clutch constructed were also carried out under various conditions, one of which consisted of running the hydraulic end at various speeds in order to determine the point at which friction surfaces engaged the

disc due to centrifugal action of oil in clutch chamber. The development of the above features was carried out in 1911 and 1912, which is particularly interesting in view of the paper, entitled "Couplings for Cruising Turbines," read before the Institution of Naval Architects in June, 1913, by Mr. H. E. Yarrow and published in "Engineering" July 4, 1913.

# TRANSMISSION OF THE PROPULSIVE POWER IN SHIPS.

By Ernest N. Janson, Member.

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The potential heat energy contained in the coal laid upon the fire grate or in the oil injected in the furnace space of a boiler converted into B.t.u., constitutes the heat supply furnished an engine or turbine steam plant.

Similarly the heat energy contained in the oil or gas furnished direct to the cylinders of an internal-combustion engine, or in the coal or other fuel in a gas producer, represents the total heat supply furnished to said types of power plants.

A very small percentage of the total B.t.u. contained in the original amount of fuel furnished for the operation is returned in mechanical equivalents as motive power. This percentage depends, of course, upon the individual efficiency of the constituent parts of which the plant as a whole is made up.

In marine work, in present-day use, such parts may be enumerated as consisting of—

1. The parts transforming the potential heat energy of coal or oil into effective heat energy of water, steam or a fluid of gas mixture.

2. The parts transforming the effective heat energy into mechanical work and rotary motion.

3. The parts transmitting the motion of the prime mover to the propeller.

To accomplish the process to be performed in No. 1, we find such apparatus as boiler, producer or apparatus for the combustion of the fuel and formation of a fluid possessed of energy necessary. In No. 2 steam engine, steam turbine or internal-combustion engine, together with all necessary auxiliaries. In No. 3 devices for transmission of the rotary mo-

tion to the propeller, either by direct coupling to engine, turbine or internal-combustion engine, or by some system of indirect transmission, such as:

*a.* Mechanical gearing.

*b.* Electrical transmission.

*c.* Hydraulic transformers.

The foregoing systems of power transmission now form important considerations in the arrangements of the propulsive machinery in marine work.

Reciprocating steam engines are universally used in direct combination with the shafting to drive the screw propeller, steam turbines, on the contrary, are used both in direct and indirect combination with the shafting as a drive for the propeller. Internal-combustion or Diesel engines are also either used in direct or indirect combination with screw propeller and shafting. With the introduction of various new types of machinery as motive power and drives for a device as sensitive as is the present-day screw propeller, with its increasing inefficiency caused by augmentation in angular velocity and surface pressure, it is only consistent with commercial as well as naval requirements and engineering interests generally to scheme out practical solutions to obtain the greatest possible *economy* with *safety* and least complication in propulsive machinery.

In pursuance of the above, therefore, numerous devices have been designed as the *best* means for the solution of this problem. It is difficult for laymen to comprehend the necessity for this "unrest" even in the field of engineering. If he, however, learns to realize the very great importance which the saving of some ounces of coal per horsepower per hour, or an increase of a few per cent. in the propulsive efficiency in, for instance, the big ships making up the fleets of navies or transatlantic lines, the reasons will be found. Extensive research and experimental work follow as a matter of course.

On the basis of 2.01 pounds of coal per H.P. of main engines of the U. S. S. *Texas* as shown on her 12-knot speed trial, the coal consumption per day of the U. S. S. *Delaware*

would be reduced from 111.7 tons to 84.98 tons and the cruising radius increased from about 7,000 to 9,300 knots (only coal being used), were the coal consumption on the *Delaware* brought down from 2.67 (from 12-knot speed trial) to that of the *Texas* (2.01 pounds) as realized on her trial. Similarly a saving of nearly 2,250 tons of coal for the round trip would result in the coal consumption by a decrease from 1.45 to 1.3 pounds of coal per horsepower, and an increase from 65 to 68 per cent. in the propulsive efficiency, in a vessel of 72,000 horsepower during a trip lasting only 5 days. The reduction in the coal bill would approximately amount to $80,000 a year at a price of three dollars a ton, counting 12 round trips.

In the following table is given names and approximate horsepower of types and combinations of propelling machinery as extensively used in nearly all classes of ships both of commercial and naval importance.

| | | | |
|---|---|---|---|
| Direct-coupled engine and propeller | ..... | 15,500,000 | Reciprocating steam engines. |
| Direct-coupled turbine and propeller | 12,000,000 | | |
| Indirectly-coupled turbine and propeller, transmission by mechanical gearing | ..... 435,000 | | |
| Indirectly-coupled turbine and propeller, transmission by hydraulic transformers | ..... 245,000 | | |
| Indirectly-coupled turbine and propeller, transmission by electric motors | ..... 12,000 | | |
| Combination machinery. — Direct-coupled reciprocating engine to propeller and direct-coupled turbine to propeller.. | 100,000 | 12,792,000 | Steam turbines. |

| | | | |
|---|---|---|---|
| Direct-coupled Diesel engine and propeller ..... | 168,000 | | |
| Indirectly-coupled Diesel engine to propeller, transmission by electric motors or gearing ..... | 2,000 | 170,000 | Internal-combustion engines. |

APPLICATIONS.

It is evident from the foregoing table, considering that less than twelve years were consumed in establishing the facts therein tabulated, that the application of the steam turbine for propulsive purposes has been more extensive and rapid than even its most sanguine supporters at any time dared to predict. It may be stated that, with the new means of transmissions, a greater field than hitherto is open for the further use and development of the turbine, especially when used for craft of moderate speed.

Owing to the high and increasing cost of oil, the steam engine and turbine will compare very favorably as an economic proposition with its comparatively new competitor, the internal-combustion engine, and will undoubtedly for some time to come hold its own, especially in very large installations where difficulties have to be overcome in the building of large cylinders as well as in the methods for their satisfactory cooling.

Means of reconciling, in a thoroughly practical manner, the high speed of revolutions of the turbine with the low speed of the propeller, required by good efficiency for both, are found in the various methods of transmitting the power, from the turbine to the propeller. The efficiency and practicability of the systems devised will play an important part in promoting and broadening the field of the marine turbine.

ENGINE COMBINATIONS FOR PROPULSION.

The following arrangements of reciprocating engines, turbines or internal-combustion engines and combinations of same, or in connection with some kind of transmission, have

received extensive and practical application for propelling ships. Only combinations with some system of transmission of the propulsive power being here considered in detail.

1. Reciprocating steam engines, direct drive of propellers.
2. Turbines, direct drive of propellers.
3. Combination reciprocating engines and turbines, each with direct drive of propellers.
4. Turbines, indirect drive of propellers, mechanical gearing.
5. Turbines, indirect drive of propellers, hydraulic transmission.
6. Turbines, indirect drive of propellers, electric transmission.
7. Internal-combustion engines, direct drive of propellers.
8. Internal-combustion engines, indirect drive of propellers, electric transmission.
9. Turbines, direct drive of propellers with combination Diesel or reciprocating steam engines for cruising purposes.

Each of the different systems of transmission previously enumerated have certain fields of application, where the inherent advantages of one or the other system render its choice preferable over the other. Thus it seems likely that the hydraulic system has a decided advantage over the mechanical gearing when the power to be transmitted is very great, owing to (perhaps) prohibitory dimensions required were gears used. Conversely the mechanical gearing over the hydraulic transformer where it becomes of practical advantage to make the speed-reduction ratio very large. Moreover the electric system may have superior advantage over either the mechanical gearing or the hydraulic system where a combination of a high-speed reduction ratio and very large powers for transmission are called for by the installation in question.

The advantages or disadvantages as given below may apply, wholy or partly, as much for one system as for the other, depending largely upon the combination with which it becomes connected.

## ADVANTAGES AND DISADVANTAGES.

*Mechanical Gearing.*—Among the principal *advantages* claimed for mechanical gearing we find the following:

*a.* Average reduction in weight of machinery of about 15 per cent. compared with straight turbine drive.

*b.* Saving in coal consumption of about 15 per cent. on same basis.

*c.* High mechanical efficiency of the gearing amounting to about 98 per cent.

*d.* Very large speed-reduction ratio permissible, possibly as high as 30 ÷ 1, or more.

*e.* Simplicity of mechanism.

Among the principal *disadvantages* admitted, we note:

*f.* Noise, although slight.

*g.* Wear, although only nominal in amount.

*h.* Limitation of power to be transmitted, owing to impracticable dimensions necessary for strength and durability.

*k.* Non-reversibility and only about 50 per cent. backing power.

*Hydraulic Transformer.*—Among the principal *advantages* claimed for hydraulic transformers are:

*a.* Reversibility and about 85 per cent. backing power.

*b.* Practically no limitation as to power to be transmitted, the turbine itself being the limiting power.

*c.* No noise, except what may be created by friction in bushings or bearings.

*d.* No wear, relatively speaking.

*f.* With turbine running always in one direction there is less chance for blade stripping.

*g.* Increased maneuvering qualities due to greater backing power.

*h.* High degree of superheat may be used (up to 200 degrees F. in turbines which are not reversed).

*k.* Increased economy and saving in weight when compared with direct turbine drives.

Among the *disadvantages* are:

*l.* Lower efficiency, about 89 per cent., when placed in comparison with mechanical gearing.

*m.* Limitation in speed-reduction ratio, the maximum being about 6 ÷ 1.

*n.* Complication in mechanism.

*Electric Transmission.—Advantages* claimed for the electric system of propulsion are:

*a.* Speed-reduction ratios may be varied by changing the motor winding.

*b.* Partial use of equipment for low powers.

*c.* Reversibility without backing turbines.

*d.* Saving of weight and added steam economy.

*f.* Transmission not limited by power.

*g.* Steady running, turbine being uninfluenced by racing of screws.

Among the *disadvantages,* the following items have been the subject of discussion among engineers. Further trials, however, are necessary for their substantiation.

*h.* Ventilation of generators.

*k.* Cooling of resistances.

*l.* Complexity of accessories.

*m.* Effect of high-tension voltage in steel ships on the ships' instruments and personnel.

For the benefit of those who have not had the opportunity to acquaint themselves with the general features of the transmission systems previously referred to, the following short descriptions are given.

### DESCRIPTION OF TRANSMISSIONS.

*Mechanical Gearing.*—Mechanical gearing for speed reduction between turbine and transmission shafts has been successfuly used in the De Laval steam turbine for more than a quarter of a century. Many engineers, and some manufacturers of other types of turbines, condemned the De Laval turbine as impractical and dangerous owing to its gearing. The introduction generally, in even very large units, of this

feature in connection with steam turbines will now sufficiently broaden the field of their application to warrant, and render thoroughly practical, their installation where formerly geared turbines were looked upon as quite inadmissible.

To meet the requirements for successful operation of mechanical gearing in connection with steam turbines, the following items must be carefully observed:

1. The gearing must be durable, implying proper tooth pressure and efficient lubrication.

2. Practically noiseless, implying correct tooth contours.

3. The transition of pressure from one tooth to another must be without shock or vibration, which is best met by using the helical type of gear.

4. Complete line contact should exist during the time the teeth are in mesh and the angle should be such as to give contact between several teeth at one time. The true involute tooth fulfills the foregoing.

5. The gear must be double, with the teeth of each cut in opposite directions.

6. The bearings for both pinion and gear must be supported so as to maintain a uniform distance between the centers.

Upon the theory of a necessity existing for flexible supports of the pinion bearings Melville and Macalpine patented a reduction gear in 1909, in which the pinion bearings are supported on a "floating frame" resting on a rigidly suspended I-beam. The lateral flexibility of this beam, it is claimed, is furnishing the adjustment required by any inequality of tooth pressure.

The principal action accomplished by the "floating frame" is claimed to enable the axis of the pinion to oscillate in a vertical plane parallel to a similar plane through the gear, the object of which was to overcome any unevenness in pressure or wear, caused by irregularities in construction, on the teeth. A splendid description of this gear will be found in "Engineering" of September 17, 1909.

Later developments in the arrangement of flexible-bearing supports resulted in a new gear being designed by the West-

inghouse Machine Company, of East Pittsburgh, Pa. Instead of oscillation of the "floating frame" being obtained by the inherent flexibility of a steel I-beam, the oscillation of the frame, which carries the pinion bearings, is obtained in the new gear by pistons floating in cylinders, built in the frame, both on the bottom for ahead rotation and on the top for astern rotation. The positions of the pistons are controlled by automatic valves, which admit or exhaust oil from the cylinders as required. A complete description of this gear will be found in Volume 24 of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.

In the original De Laval gearing, as well as in Parsons mechanical gearing fitted on various vessels, absolutely rigid supports for pinion and gear bearings were and are fitted. The theory upon which this is being done is that no flexible contrivance would adjust itself in time to inequalities, produced in the tooth pressure by irregularities of tooth contour or pitch, owing to the irresponsive nature of the masses involved. "Prevention is better than cure," prevention being accomplished by accuracy in manufacture. To insure absolute accuracy in cutting of gears and pinions extensive and costly experiments have been carried out both in the designing and the building of the gear-cutting machines. Any corrective finishing process or hand work, after the gear once is cut, being considered entirely inadmissible.

A process for making gearing, as used in connection with turbines of Parsons type, is described in a paper read by Sir Charles A. Parsons before the Institution of Naval Architects in 1913 (see *Transactions* of 1913). The general and distinguishing features of this gearing consist of (as in the De Laval gear to which reference has been made) helical teeth with involute contour, twin-gear and pinion mounted on inflexible bearing supports. Accuracy in the cutting of the teeth, together with the natural elasticity of the bearing and the connection of pinion and turbine shafts by a flexible coupling have resulted in a sufficiently satisfactory and practically noiseless and durable gearing.

The material used in the gearing fitted in ships with Parsons arrangement is:

Pinion. { Nickel steel:
Tensile strength, 90,000 pounds per square inch.
Elastic limit, 55,000 pounds per square inch.
Elongation, 25 per cent.

Wheels. { Carbon steel:
Tensile strength, 70,000 pounds per square inch.
Elongation, 26 per cent.

*The Föttinger Hydraulic Transformer.*—The Föttinger hydraulic transformer was tried experimentally in 1909 by the Hamburg-American Steamship Line to ascertain its value as a speed-reduction gear for marine purposes in connection with steam turbines. It is now being applied in large and small powers in various classes of ships and has proved itself successful both mechanically and economically. The latest installation of this gear, at the time of this writing, will be found in the Hamburg-American Channel T. S. S. *Königin Luise.* Elementarily described it consists of two centrifugal-pump wheels keyed to the primary shaft, which also constitutes the turbine shaft. On the secondary, or propeller shaft, are mounted three impulse-reaction water-turbine wheels, the after two for giving ahead motion and the forward one for backing motion. The part for ahead motion absorbs the energy delivered to it from the water discharged by the pump wheel in two stages, the primary stage being part of the backing water-turbine wheel. Having given up part of its energy in the primary ahead stage, the water moves in a continuous compact flow through guide vanes fastened in the casing to the second-stage wheel, there giving up the remainder of its energy. The discharge from this wheel flows back in the pump wheel giving up to it any velocity yet remaining. The first and second-stage wheels for ahead motion are bolted together, at the same time forming part of the backing wheel placed in the forward wheel stage. The backing wheel receives its energy from the water discharged by the forward

pump wheel guided into it by the stationary vanes fastened in the casing. The vanes in the backing and ahead turbine wheels are spaced in opposite directions to one another thereby givng reversed motion. By means of a change valve, controlled by a lever, water may be admitted from either pump wheel to the corresponding turbine wheel giving the desired direction of motion for ahead or back. An exceedingly short space of time is required for reversing the motion.

The feed water is used in series with the transformer, and during its passage through same will gain considerable heat owing to the work done upon it by friction, the loss through which is therefore partly regained by the plant in heat. Because of the higher velocities creating increased fluid friction, the efficiency of the transformer decreases with an increase in speed reduction ratio.

The transformer wheels and vanes are made of bronze and the casings of cast iron.

*Electrical Transmission.*—Generally speaking, the electrical apparatus for propulsive purposes in marine installations consists of turbine-driven generating units, induction motors, with or without groups of water-cooled resistances for reversing purposes and suitable instruments and switching devices. Besides the foregoing, and operating in connection with same, is found the motor driving the generator, either a steam turbine, Diesel or other type of engine, and, in the event of a steam turbine or engine, of course, the steam-generating plant.

At the present time there are two distinct systems in successful operation, one devised by Mr. W. L. R. Emmet, of the General Electric Electric Company, Schenectady, N. Y., as installed in the U. S. Collier *Jupiter,* and the other, called the Mavor and Coulson system, as installed in the S. S. *Tynemount,* built in England for traffic on the Great Lakes of North America.

By Mr. Emmet's system, where the installation is made for ships requiring to run at different speeds for long periods of time, as, for instance, battleships, the motors are arranged for pole changing in such a way that the stator can be connected,

say, for either 30 or 50 poles, the 30-pole connection being used for high-speed operation and the 50-pole connection for operation at cruising speeds. Where the equipment is proposed for ships running practically at normal speed for the entire time no arrangements are provided for pole changing. For description of installation of U. S. *Jupiter* see Volume 25, No. 4, JOURNAL AMERICAN SOCIETY NAVAL ENGINEERS.

The Mavor system is based on a spinner motor, which gives three speeds in either direction at maximum motor efficiency, the control being had from two double-throw switches. The spinner is arranged concentric with the rotor of an ordinary squirrel-cage motor. The winding on the inside of the spinner serves to drive the rotor, while on its outside periphery the spinner has a squirrel-cage winding that enables it, in turn, to be driven as a motor by a surrounding stator. The spinner is provided with a brake and may be held while the rotor is revolving, and its speed may, in this manner, be either added to or subtracted from that of the rotor, giving three speeds in either direction without the use of resistances in the rotor circuit and without pole changing. An equipment of this kind would weigh considerably more than an all-turbine drive. (See JOURNAL OF AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. XXII, an article on Naval Engineering Progress, by H. I. Cone, U. S. N., read before the U. S. Naval War College.)

## INSTALLATIONS.

In Table II are enumerated a few ships in which are installed various systems for the transmission of the propulsive power either for full power or in part for cruising purposes.

Among recent noteworthy installations are:

U. S. Collier *Jupiter* with electric transmission furnished by the General Electric Company of Schenectady, N. Y., which vessel recently has concluded satisfactory trials and exceeded the guarantee both as to speed and economy of steam.

The Hamburg-American T. S. *Königin Luise* with Föt-

TABLE II.

| Name of ship or tonnage. | Year tried. | System of power transmission. | Prime mover. | Maximum speed, in knots per hour. | Revolutions of propeller per minute. | Speed reduction ratio. | Total shaft horse-power, S.H.P. | Water consumption, pounds per S.H.P. | Remarks. |
|---|---|---|---|---|---|---|---|---|---|
| Cargo vessel *Vespacian*..... | 1910 | Mechanical gearing (Parsons). | 1 H.P. turbine .....<br>1 L.P. turbine.<br>Parsons. | 10.34 | 77 | 20 ÷ 1..... | 945 | Main turbine, 14.81. | 1 screw. |
| Channel steamers *Normannia* and *Hantonia*. | 1911 | .....do ..... | 2 H.P. turbines.....<br>2 L.P. turbines.<br>Parsons. | 19.7 | 310 | 6.4 ÷ 1 h.p.<br>4.4 ÷ 1 l.p. | 4,980 | Main turbine, 12.3. | 2 screws. |
| Cargo steamer *Cairncross*... | 1912 | .....do ..... | Same ..... | 10.2 | 61.76 | 26.2 ÷ 1 ..... | 1,590 | Main turbine, 14.0. | 2 screws. |
| Channel steamer *Paris*..... | 1913 | .....do ..... | 2 H.P. turbines.....<br>2 L.P. turbines.<br>Parsons. | 25.07 | 435 | 4.25 ÷ 1....<br>6.0 ÷ 1. | 14,000 | ..... | 2 screws. |
| U. S. S. *Melville* ..... | Building | Mechanical gearing (Westinghouse). | 1 H.P. turbine.....<br>1 L.P. turbine.<br>Parsons. | ? | 110 | 18 ÷ 1..... | 4,000 | ..... | 1 screw. |
| U. S. S. *Bushnell*..... | Do. | Mechanical gearing... | H.P. turbine .....<br>1 L.P. turbine.<br>Parsons. | 14 | 125 | 16 ÷ 1..... | 2,500 | ..... | 1 screw. |
| Hamburg American Line T. S. S. *Koenigin Luise*. | 1913 | Hydraulic transformers (Föttinger). | 2 A.E.G. turbines ..... | 20.05 | 453 | 4.03 ÷ 1.... | 5,330 | Main turbine, 12.0. | 2 screws.<br>Sup. 70 degrees F. |
| U. S. Collier *Jupiter*..... | 1913 | Electric transmission (Emmet). | 1 Curtis turbine ..... | 14.78* | 114.33 | 18.6 ÷ 1.... | 6,435 | Main turbine, 11.45. | 2 screws. |
| Cargo vessel *Tynemount*.... | 1913 | Electric transmission (Mavor). | Two 6-cyl. Diesel engines. | 14.78 | 78 | 5.1 ÷ 1..... | 500 | ..... | 1 screw.<br>1 Ind. Motor. |
| U. S. S. *Cushing*..... | Building. | Mechanical gearing (Falk). | ..... | ..... | 250 | 8 ÷ 1..... | ... | ..... | For cruising purposes only.<br>2 screws. |
| U. S. S. *Pennsylvania*..... | Do. | Mechanical gearing (Westinghouse). | 2 H.P. turbines.<br>2 L.P. turbines.<br>2 Astern turbines.<br>4 Cruising turbines on 2 shafts.<br>Curtis. | ..... | ... | ... | ... | ..... | For cruising purposes only.<br>4 screws. |

* Later trials gave 15.1 knots for 48 hours on 115.7 R.p.m. and 6,940 H.P.

tinger hydraulic transformer which has given general satisfaction both as regards practicability and economy.

A large number of steamers equipped with mechanical gearing by their successful operation have proved the value of this method of power transmission.

In an arrangement of machinery of the new Hamburg-American liner *Imperator,* if said steamer were equipped with hydraulic transformers, a saving of about 35 per cent. in weight of machinery and 33 per cent. in space is claimed. Backing power would be 85 per cent. against 55 per cent. as now with direct turbine drive, besides a considerable reduction in coal consumption.

The S. S. *Cairncross,* fitted with Parsons turbines and mechanical gearing, but equipped in every other respect like her sister boat, the S. S. *Cairngowan,* fitted with reciprocating triple-expansion engines, and tried under identically the same conditions, showed 14.93 per cent. better coal economy than the *Cairngowan.* However, the design of the *Cairngowan's* engines was not the best represented by late reciprocating-engine practice as far as economy is concerned.

## EFFICIENCY.

The only acceptable and reliable manner by which to make comparison of merits possessed by different mechanical devices is to try them out in actual operation and thereby to establish efficiency data. But as it is hardly ever possible to so run ships, especially when controlled by separated interests, to obtain comparative data under exactly similar conditions for all the different components of the machinery installation, there is yet usually sufficient data obtainable upon which to base general results. It is with this in view that the following investigation has been made. Only machinery in which steam forms the generating power is here considered.

Looked upon in its simplest form, we may say the machinery installation of a steam plant in a vessel consists of the following parts:

1. The steam-generating plant.
2. The propelling engines.
3. The auxiliaries.
4. The device for transmission of the power between propelling engine and propeller.
5. The propeller.

It is well known that none of these component parts, in any sense of the word, approach perfection, but that losses occur at every stage at the expense of energy. To analyze the losses occurring in each of the individual parts would be an endless task and is not necessary for our purpose. We may at once concede that in the present-day modern ship but slight differences need exist in the general efficiency of the steam-generating plant, in the auxiliaries and in the propeller, when conditions of speed and pressure for giving adequate efficiency of the propeller have been properly taken into consideration. Thus the problem reverts back to a determination of efficiency of the propelling engines and the expenditure of heat energy to obtain the necessary amount of thrust to drive the ship at its stipulated speed. In regard to the propelling engines, the conditions under which the heat in the steam is made available and the speed with which they are run are of principal consideration, while in the determination of the expenditure of heat energy necessary for thrust the efficiency of the steam-generating plant and the device for the transmission of the power as well as the propeller, all play important parts.

It is with respect to the "thermal efficiency" of the propelling engine that the transmission device becomes of very superior importance. When running in connection with the steam turbine this efficiency has been materially improved, as shown by the decreased steam consumption. This is principally due to the higher vane speed attained in turbine run in connection with gearing of some kind.

A perfect engine converting all the heat energy in the steam into work would require 2,545 British thermal units per horsepower per hour, or the equivalent of 1,980,000 foot pounds. Divided by the number of heat units per hour per horsepower

actually consumed by the engine gives the "thermal efficiency" of the engine. Thus we may write:

Thermal efficiency of engine =

$$\frac{\text{Energy of one horsepower hour expressed in B.t.u. } (= 2{,}545)}{\text{B.t.u. required per horsepower hour of actual engine.}}$$

Thus if an engine uses "$p$" pounds of steam per hour per horsepower and "$b$" thermal units for each pound of steam (which is equivalent to total heat in steam at engine pressure less heat returned), then the engine uses $p \times b$ thermal units per hour per horsepower and its "thermal efficiency" $= \frac{2{,}545}{p \times b}$.

The value thus obtained for the "thermal efficiency" will depend upon whether the horsepower of the engine corresponds to the indicated work done by the steam in the cylinders or to that actually transmitted to the shaft as given by a brake or torsion meter. In the former case we deal with indicated horsepower and in the latter with shaft horsepower. But the shaft horsepower is always less than the indicated, due to the power absorbed by friction, and, therefore, on the basis of total steam consumption, the steam consumption, per shaft horsepower will be greater than when dealing with indicated horsepower. In order to have an approximately true value of comparison for the different engines given in the examples, the steam consumption in all cases is based on shaft horsepower and one shaft horsepower is taken equal to .92 indicated horsepower. Thus if "$e$" denotes the "thermal efficiency" arrived at from steam consumption per shaft horsepower the same efficiency will be $.92 \times e$ when the steam consumption is obtained on the basis of indicated horsepower. The examples which follow for several types of the most modern marine installations will illustrate in certain measure their economic value. A resumé of this work is given in Table III.

The figures presented in the examples can not be claimed as absolutely correct, as in some cases an allowance had to be made for the steam consumption of the auxiliaries, while in other cases the estimated steam consumption per shaft horse-

TABLE III.—GENERAL DATA AND EFFICIENCIES.

| Name of ship. | Steam pressure at throttle, pounds gage. | Vacuum, inches mercury. | H.P. | Lbs. water per hour per H.P., all machinery. | Pounds coal or oil per hour per H.P. | B.t.u. per pound, coal or oil. | R.p.m. of propeller. | Speed, in knots. | Thermal efficiency, main engines, per cent. | B.t.u. per minute per S.H.P. of main engines. | B.t.u. per minute used by whole machine installation per lb. indicated thrust. | System of power transmission. | Remarks. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| U. S. S. *Texas*..... | 271.1 | 24.24* | 28,373 | 15.06 | 1.73 | 14,970 | 124.56 | 21.05 | 16.03 | 264 | 34.24 | Direct-connected shafting, twin screws. | Full power; coal burning, 13.25 lbs. net per I.H.P. main engines. |
| U. S. S. *Texas*..... | 277.7 | 26.2* | 4,640 | 17.37 | 2.01 | 14,970 | 69.26 | 12.13 | 16.08 | 287 | 20.2 | Direct-connected shafting, twin screws. | Low power; 35° superheat 13.9 lbs. net per I.H.P. main engines. |
| U. S. collier *Jupiter*. | 164.0 | 28.95 | 6,435 | 12.72 | 1.74 | 13,900† | 114.3 | 14.78 | 19.4 | 220 | 18.3 | Electric transmission, twin screws. | Full power; coal burning, 11.42 lbs. net per S.H.P. |
| U. S. collier *Jupiter*. | 180.6 | 28.95 | 1,855 | 15.02 | 2.42 | 13,909 | 76.8 | 9.93 | 17.3 | 244 | 17.1 | Electric transmission, twin screws. | Low power; 12.77 pounds net per S.H.P. |
| U. S. T. B. D. *Cassin*. | 244.4 | 27.8 | 15,307 | 14.93 | 1.185 | 19,300 | 60.2 | 30.13 | 17.8 | 237 | 42.6 | Direct turbine connection. Reciprocating engines for cruising. | Twin-screw, oil burning, full power. |
| U. S. T. B. D. *Cassin*. | 205.0 | 27.6 | 372 | 38.71 | 4.34 | 19,300 | 197.3 | 12.0 | 5.9 | 840 | 50.6 | Direct turbine connection. Reciprocating engines for cruising. | Twin-screw, oil-burning, low power. |
| Hamb. Am. T. S. S. *Koenigin Luise*. | 240.0 | 28.27 | 5,330 | 12.00‡ | 1.31 | 13,500 | 453.3 | 20.05 | 17.9 | 236 | 19.9 | Föttinger hydraulic transmission; twin screws. | Full power; 70° F. sup. |
| Channel T. S. S. *Normannia*. | 160.0 | 27.6 | 5,000 | 14.3 | 1.34 | 13,500 | 310.0 | 19.68 | 17.73 | 224.2 | 21.6 | Parsons' mechanical gearing; twin screws. | Full-power; 12 lbs. net per S.H.P. main engines. |

* L.P. cylinder. † Pocahontas and New River, estimated. ‡ Main engines only.

power of main turbines is given. Where not otherwise stated 15 per cent. of the total steam used at full power and 20 per cent. at low power is assumed consumed by the auxiliaries.

U. S. S. "TEXAS."

*Data.*

| | |
|---|---|
| I.H.P., main engines, full power | 28,373 |
| Indicated thrust, pounds | 357,000 |
| Weight of steam passing through cylinders, lbs., per min. | 6,265* |
| I.P. cylinder jacket, lbs., per min. | 8.52** |
| Temperature of steam at throttle valve, degrees F | 412† |
| in the exhaust at cylinder, degrees F | 139† |
| Pounds of water per hour all machinery per I.H.P. main engines | 15.06 |

*Result.*

| | |
|---|---|
| Total B.t.u. in coal used per minute for whole machinery installation $= \frac{1.73 \times 28{,}373 \times 14{,}970}{60}$ | 12,245,000 |
| Total heat of 6,265 pounds steam passing through cylinders per minute (1,207.6 × 6,265) | 7,564,000 |
| Less heat returnable in the feed from exhaust (106.8 × 6,265) | 669,000. |
| Total heat of 8.52 pounds of steam at boiler pressure passing through I.P. cylinder jacket per minute | 10,300 |
| Less heat returned from jacket (at 240 degrees F) | 1,770 |
| Net heat supply to engine | 6,903,530 |
| Thermal efficiency $= \frac{2.545}{(1{,}207.6 - 106.8)\ 13.25} \times .92$ per cent | 16.03 |
| B.t.u. per minute per I.H.P. $= \frac{6{,}903{,}530}{28{,}373}$ | 243 |
| S.H.P. | 264 |
| B.t.u. per minute per pound of indicated thrust $= \frac{12{,}245{,}000}{357{,}000}$ | 34.24 |

*Data.*

| | |
|---|---|
| I.H.P. main engines, cruising speed | 4,640 |
| Indicated thrust, pounds | 115,000 |
| Weight of steam passing through cylinders, lbs. per minute | 1,075 |
| I.P. cylinder jacket, lbs. per minute | 8.52* |
| Temperature of steam at throttle valve, degrees F. (35° F. Sup.) | 450 |
| Temperature of steam in the exhaust at cylinder, degrees F. | 124 |
| Pounds of water per hour all machinery per I.H.P. main engines | 17.37 |

*12 per cent allowed for auxiliaries.
**Estimated.
†Estimated from pressure.

*Result.*

| | |
|---|---|
| Total B.t.u. in coal used per minute for whole machinery installation $= \frac{2.01 \times 4,640 \times 14,970}{60}$ | 2,327,000 |
| Total heat of 1,075 pounds steam passing through cylinders per minute (1,230 × 1,075) | 1,322,250 |
| Less heat returnable in the feed from exhaust (91.9 × 1,075) | 98,800 |
| Total heat of 8.52 pounds of steam at 76.8 lbs. pressure passing through I.P. cylinder jacket per minute | 10,030 |
| Less heat returned from jacket (at 194 degrees F) | 1,382 |
| Net heat supply to engine | 1,232,100 |
| Thermal efficiency $= \frac{2,545}{(1,230 - 91.9)\ 13.9} \times .92$ per cent | 16.08 |
| B.t.u. per minute per I.H.P. $= \frac{1,232,100}{4,640}$ | 265 |
| S.H.P. | 287 |
| B.t.u. per minute per pound of indicated thrust $= \frac{2,327,000}{115,000}$ | 20.2 |

U. S. Collier "Jupiter."

*Data.*

| | |
|---|---|
| S.H.P., main turbine, high power | 6,435 |
| Indicated thrust, pounds $= \frac{\text{S.H.P. thrust}}{.92}$ | 141,600 |
| Weight of steam passing through turbine, lbs. per minute | 1,234* |
| Temperature of steam at throttle valve, degrees F | 372 |
| in the exhaust at condenser, degrees F. | 80.5 |

*Result.*

| | |
|---|---|
| Total B.t.u. in coal used per minute for whole machinery installation $= \frac{1.74 \times 13,900 \times 6,435}{60}$ | 2,594,000 |
| Total heat of 1,234 pounds steam passing through turbine per minute (1,196 × 1,234) | 1,475,864 |
| Less heat returnable in feed from exhaust (48.5 × 1,234) | 59,770 |
| Net heat supply to turbine | 1,416,094 |
| Thermal efficiency $= \frac{2,545}{11.42\ (1,195.5 - 48.5)}$, per cent | 19.4 |
| B.t.u. per minute per S.H.P. | 220 |
| B.t.u. per pound per minute of indicated thrust | 18.3 |

*Data.*

| | |
|---|---|
| S.H.P., main turbine, low speed | 1,855 |
| Indicated thrust, pounds | 60,800 |
| Weight of steam passing through turbine, lbs. per minute | 395† |

*Ten per cent of steam deducted for auxiliaries from total used for all machinery. 13,900 B.t.u. assumed as heat equivalent of coal used.

†Fifteen per cent. of steam deducted for auxiliaries from total used for all machinery.

| | |
|---|---|
| Temperature at throttle valve, degrees F. | 379 |
| in the exhaust, degrees F. | 80.5 |

*Result.*

| | |
|---|---|
| Total B.t.u. in coal used per minute for whole machinery installation $= \frac{2.42 \times 13,900 \times 1,855}{60}$ | 1,040,000 |
| Total heat of 395 pounds of steam passing through turbine per minute (395 × 1,197.5) | 472,900 |
| Less heat returnable in feed from exhaust (48.5 × 395) | 19,160 |
| Net heat supply to turbine | 453,746 |
| Thermal efficiency $= \frac{2,545}{12.77\ (1,197.5 - 48.5)}$, per cent | 17.3 |
| B.t.u. per minute per S.H.P. $= \frac{453,740}{1,855}$ | 244 |
| B.t.u. per minute per pound of indicated thrust | 17.1 |

U. S. T. B. D. "CASSIN."

*Data.*

| | |
|---|---|
| S.H.P., main turbine, full power | 15,307 |
| Indicated thrust, pounds | 137,000 |
| Weight of steam passing through turbines, in lbs. per min. | 3,240 |
| Temperature at throttle valve, degrees F. | 404 |
| in the exhaust, degrees F. | 104 |

*Result.*

| | |
|---|---|
| Total B.t.u. in oil used per minute for whole machinery installment | 5,834,000 |
| Total heat of 3,240 pounds of steam passing through turbines | 3,894,480 |
| Less heat returnable in the feed from exhaust | 264,600 |
| Net heat supply | 3,629,880 |
| B.t.u. per minute per S.H.P. $= \frac{3,629,880}{15,307}$ | 237 |
| pound indicated thrust $= \frac{5,834,000}{137,000}$ | 42.6 |
| Thermal efficiency $= \frac{2,545}{(1,205 - 81.6)\ 12.7}$, per cent | 17.8 |

*Data.*

| | |
|---|---|
| S.H.P., main turbines, cruising speed | 372 |
| Indicated thrust, pounds | 10,250 |
| Weight of steam passing through cruising engine and turbines, pounds per minute | 280 |
| Temperature at throttle valve, degrees F. | 386 |
| in the exhaust, degrees F. | 107.5 |

*Result.*

| | |
|---|---|
| Total B.t.u. in coal used per minute for whole machinery installation | 519,000 |
| Total heat of 280 pounds of steam passing through cruising engine and turbines per minute (1,199.7 × 280) | 336,000 |
| Less heat returnable in feed from exhaust (74.95 × 280) | 21,000 |
| Net heat supply to engine and turbines | 315,000 |
| Thermal efficiency $= \frac{2,545}{38.71\ (1,199.6 - 74.95)}$, per cent | 5.9 |
| B.t.u. per minute per S.H.P. $= \frac{315,000}{372}$ | 840 |
| Per minute per pound of indicated thrust $= \frac{519,000}{10,250}$ | 50.6 |

"KÖNIGIN LUISE."

*Data.*

| | |
|---|---|
| S.H.P. | 5,330 |
| Indicated thrust, pounds | 78,900 |
| Weight of steam in pounds passing through turbine per min. | 1,066* |
| Temperature at throttle valve, degrees F | 472† |
| in the exhaust, degrees F | 96.6 |

*Result.*

| | |
|---|---|
| Total B.t.u. in coal used per minute for whole machinery installment | 1,570,000 |
| Total heat of 1,066 pounds of steam passing through turbines | 1,330,000 |
| Less heat returnable in the feed from the exhaust | 68,800 |
| Net heat supply | 1,261,200 |
| B.t.u. per minute per S.H.P. $= \frac{1,261,200}{5,330}$ | 236 |
| B.t.u. per minute per pound indicated thrust $= \frac{1,570,000}{78,900}$ | 19.9 |
| Thermal efficiency $= \frac{2,545}{12\ (1,247 - 64.5)}$ | 17.9 |

"NORMANNIA."

| | |
|---|---|
| S.H.P., full power | 5,000 |
| Steam pressure, pounds | 160 |
| Speed, knots | 19.7 |
| Coal per S.H.P., pounds per hour (all purposes) | 1.34 |
| Water per S.H.P., pounds per hour (all purposes) | 14.3 |
| auxiliaries | 2.3 |
| Net water per S.H.P., pounds per hour, main engines | 12 |

*Estimated for turbines and transformers.
†70° F. superheat.

| | |
|---|---|
| Revolutions per minute | 310 |
| Pitch of screw (estimated) | 8′0″ |
| Indicated thrust, pounds $= \frac{5{,}000 \times 33{,}000}{8 \times .92 \times 310}$ | 72,300 |
| Total B.t.u. in coal used per minute for whole installation $= \frac{1.34 \times 5{,}000 \times 13{,}500^*}{60} =$ | 1,507,500 |
| Weight of steam passing turbines per minute (net 12 lbs.) $= \frac{12 \times 5{,}000}{60} =$ lbs. | 1,000 |
| Total heat in 1,000 lbs. steam passing through turbines | 1,196,000 |
| Less heat in exhaust (vacuum assumed, 27.6″) | 74,950 |
| Net heat supplied | 1,121,050 |
| Heat used per S.H.P. per min. $= \frac{1{,}121{,}050}{5{,}000} =$ B.t.u. | 224.2 |
| B.t.u. per pound thrust $= \frac{1{,}507{,}500}{72{,}300} =$ | 21.6 |
| Thermal efficiency, full power $= \frac{2{,}545}{12 \times 1{,}196} =$ per cent | 17.73 |

*Heat equivalent of coal (assumed) B.t.u., 13,500.

# NOTES.

## THE DIESEL ENGINE.

PAPER PREPARED BY MR. CARELS, WITH DISCUSSIONS AND OTHER NOTES ON OIL ENGINES.

Mr. Carels' paper is reprinted from "Engineering," discussion from other sources has been added, together with some notes by LIEUT. C. N. HINKAMP, U. S. N.

### PRESENT POSITION OF THE DIESEL ENGINE, CHIEFLY IN MARINE PROPULSION.*

BY GEORGES CARELS.

We have no intention of establishing here a complete history of the Diesel engine or of describing minutely the numerous types that have been successively put on the market, by a steadily increasing number of firms; you are kept well informed by the English technical press, and we have only words of praise for the very impartial and clear judgment of which they have constantly given proof.

It cannot be denied that Diesel-engine development is not progressing today as rapidly as was expected a year or two ago. And what are the reasons of this momentary relaxation? It is quite certain that the present prohibitive prices of liquid fuel are responsible for the situation to a large extent, but there are other reasons as well.

We have just praised the English technical press, but we cannot do the same for the continental press in its entirety. Their information has been often misleading. Optimists have seen in the Diesel a universal engine, called upon to supersede in the near future all other types of prime movers on land and at sea. And now that their predictions are not immediately realized they have grown very quiet and find no defence for their opinions of yesterday. All this contributes towards diminishing public confidence in Diesel engines.

In the early days of the marine Diesel engine everything pointed towards the high-speed engine as the engine of the future. Small wonder, then, that many firms took up the manufacture of this type first. And the temptation of trying to secure the first Admiralty orders, even with the risk of insufficient time for extensive preliminary experiments, was very great, because Admiralties are good customers when satisfied, as we all know. It should have been better borne in mind that—

(1) At that time there was no sufficient proof of the reliability of high-speed Diesels under every condition of load and drive.

(2) That slow-speed two-stroke motors having proved a success and high-speed four-strokes possible, was no sufficient reason for concluding that high-speed two-strokes were as easy to build.

(3) That for powers under 200 horsepower, it was not possible with either a reversible two or four-stroke multi-cylinder Diesel to compete successfully with cheaper, more easily handled, and more robust types of less cylinders, maneuvered by means of a clutch or a reversible propeller.

(4) That it is not sufficient to build an engine that reverses well on the test bed under no load to make it a desirable engine from the marine point of view.

*Northeast-Coast Institution of Engineers and Shipbuilders.

(5) That it is a very unsafe course to follow to advertise a new type of engine before it has given sufficient proof of its quality; in fact, extremely unsafe for a firm that has but just started the manufacture of Diesel engines and has most of its engines at the best on paper only.

(6) That there is no sense from a commercial point of view in building a high-speed two-stroke engine as heavy or heavier than its four-stroke rival—because of very low mean effective pressure—unless it is considerably simpler and more reliable than the four stroke.

(7) That the characteristic feature of a submarine engine should be extreme simplicity, great accessibility of all the essential functions, and absolute reliability rather than excessive lightness.

(8) Last, but not least, that success is not ensured by securing good Diesel-engine designers, success being far more a question of manufacture and metallurgy than some people suspect, and that experience in the right material to use, wear of cylinder liners, and clearance of pistons, is essential.

Nevertheless, this very active period from 1908 to 1912 has in many ways strongly influenced design. Neither do we assert that all marine high-speed engines created in that period are without value; very fine results have been obtained in some cases, considering the short time of development. But we do not look upon these designs, specially in the important class of submarine engines, as final. Most have some trouble in fulfilling their guarantees, and all those who have seen and manipulated these engines will agree that few of them give an impression of absolute security, and that it takes strong nerves to withstand the strain produced by their quite infernal rattle.*

*The feature spoken of here, namely, the noise of the Diesels, is one of the greatest disadvantages of this type of engine, and the elimination of this feature is demanding the attention of designers now more than ever beore. Extreme simplicity is the solution, which demands the reduction in number of the valves and actuating mechanism.

So many failures and difficulties have had the distinct advantage—at very considerable expense, it is true—of showing how a Diesel should *not* be made; the numerous papers and publications and discussions on the subject have served to clear many difficulties. Now, a sounder policy seems to be establishing itself. There is a distinct return to more moderate speeds, more accessible types, than the continental high-speed engine. Firms are limiting their activity to fewer types, apparently aware that the fewer they are the better and more rational their manufacture.

The real requirements of the market are being studied more seriously and the necessity for more exact experimental work is being recognized. Indeed, it should be always borne in mind that the original Diesel engine greatly owed its astounding success to the four years of minute laboratory work that preceded its appearance. Many firms have bought their present experience at a very high price that might have been very much smaller had they started from the firm basis of sufficient exhaustive preliminary tests on experimental engines. Inventive ability, although essential in a progressive factory, must be channelized, held down to certain sound principles based on good common sense. It should not be allowed to invade the designing department, with which it has nothing in common. Designing must always be considered from a manufacturing point of view.

We will now briefly summarise the teachings of what we have just laid down, more especially concerning marine engines, comparing first high-speed two-stroke with four-stroke engines.

(1) The two-stroke is safer because it has no exhaust valves.

(2) Its reversing gear is simpler than that of the four-stroke and considerably cheaper.

(3) Comparing engines of same power,† the two-stroke has a more

†And same number of cylinders.—ED. THE E.

equal turning moment.

(4) But the mean temperature of its cycle is higher, rendering therefore piston cooling necessary sooner than with four strokes.

(5) It consumes considerably more fuel and lubricating oil than the four-stroke.

(6) Its mean effective pressure is low, therefore a high-speed two-stroke is not necessarily lighter than a high-speed four-stroke.

(7) As a valveless engine it is still less economical from the fuel and oil point of view than the two-stroke with valves, but has the advantage of very great simplicity.

Comparing slow-speed engines of both cycles reverses the case. Here it is possible to obtain a mean effective pressure equal to or higher than that of the four-stroke, therefore the slow-speed two-stroke can be made lighter and cheaper than the four-stroke of same power and speed. The fuel consumption is, in a two-stroke, a function of the speed of the engine. At slow speed it is only about 10 per cent. higher than in the corresponding four-strokes; the same remark applies approximately to the lubricating-oil consumption. Points (1), (2), (3) and (4) still apply.

Further sub-dividing two-stroke engines into—

(1) Engines with scavenging valves in the cylinder head.

(2) Engines with no scavenging valves but with scavenging ports in the cylinder—valveless engines—we recognize that:—

For several reasons the scavenging in the last-mentioned class is less perfect than in the first. The results are a lower mean effective pressure and a higher fuel consumption.

The valveless engine is therefore not an economical engine from the fuel and weight points of view, but it has the big advantage of greater simplicity. Its running can be very smooth and noiseless; its valve and reversing gear very simple and cheap. It is easy to see that whereas the necessary volume of scavenging air increases with the square of the cylinder bore, the available breadth of the scavenging ports varies only with the first power of this bore. It is therefore necessary to bear this fact well in mind when choosing the number of revolutions of such an engine. In short, the efficiency of a high-speed two-stroke is a function of the number of revolutions to a much higher extent than in a four-stroke. Here the time necessary for combustion is not the leading factor, but rather the time necessary for sufficient scavenging.

The above remarks point out the following:

In all cases where simplicity and robustness, combined with maximum reliability, are the essential factors, and where the fuel bill is of secondary importance, we would recommend the use of the valveless two-stroke. In all cases where the fuel economy is the essential factor, but high speeds necessary because of space and weight limitations we recommend the four-stroke—electric-lighting sets for warships, &c. For all high-powered slow-speed engines for land and marine purposes—say, from 1,000 horsepower upwards—the two-stroke. This is the real domain of the two-stroke, where its advantages can be fully brought out. We will discuss this question more thoroughly further on in dealing with marine engines.

For submarine engines we are of the opinion that none of the types that have been created till now are entirely suitable. We see in the valveless engine the ideal submarine engine, but the mechanical efficiency of this type augmenting rapidly with a reduced number of revolutions, we would recommend a considerable reduction in speed. We surely hope that Admiralties are gradually coming to the conclusion that it is better to build the boat around the engine than to trust to competition between several firms to produce an adequate type for a given boat. And why should it not be possible to seek economy in weight more in other parts of submarine installation and machinery than precisely in the most essential factor where quick and smooth running is the condition *sine qua non?*

As regards marine Diesel engines, we would mention the following advantages:

(1) The efficiency of a Diesel engine is largely independent of the men working it.

(2) Further, the average fuel consumption is only slightly higher in actual work than at official trials. Professor Hoeltze reports this increase to be on an average about 14 per cent., whereas for steam engines, the average difference is 102 per cent.

Such figures should go far towards destroying the false idea that the Diesel engine is a hard engine to handle; they are absolute proof of the contrary.

(3) Moreover, the mechanical efficiency is much more independent of the load than in any other type of prime mover.

(4) The engine starts at a few seconds' notice without any previous preparations whatever. On the other hand, as soon as the ship stops the fuel consumption also stops. There are no stand-by losses. These several features bring us to the conclusion that, even considering the relative prices of coal and oil fuel, it is quite possible that the Diesel fuel bill may attain a lower figure on an average than in a steamship.

(5) More space is available on board for cargo.

(6) Fuel can be more quickly and much more easily stored, especially at sea.

(7) As compared with coal, fuel for four times the distance can be carried in a given space.

The general principles to be clearly laid down and consistently followed in designed marine Diesel engines are:

*Reliability.*—It is evident that the first and most essential condition is absolute reliability and security. A breakdown at sea may signify complete destruction of the vessel and more—loss of human life. If it is essential that our land generating sets be trustworthy, it is doubly so in marine engines.

All, even most remote, and improbable causes of breakdown must be sought for and rendered impossible; in face of the all-importance of security every other feature, such as weight, price, accessibility must be subordinated to this one factor. It is the condition *sine quâ non* of success of the new industry that public confidence be not shaken by accidents and breakdowns.

With this principle as a constant guide the designer in his research of suitable designs will naturally incline towards very massive structures, because weight is often, though not always, synonymous with strength and stiffness. But he will have to sub-divide the different parts of a Diesel into three distinct groups.

In the first group are those parts that are subjected to mechanical stresses only, but to no high temperatures. Their dimensions can be established clearly by calculation, their material with today's knowledge of metals, and their fatigue under stress safely determined; in fact, they are best established by a firm with marine experience, for they can differ only in necessary section from standard steam-engine practice.* It is here

*Some exception to this may be taken. It is believed that the author fails to consider the difference in the method of application of the load. In the steam engine the load is a gradual one; in the Diesel the load is an impact of no mean violence, in spite of the fact that the principle is that of a slow burning of the fuel in the cylinder. It is the compression and firing pressures that have to be considered, not the mean effective pressure, as that is but a small fraction of the former.

sufficient not to economize in weight and dimensions.

The crank shaft, the connecting rods, the bed plate, and the cylinder columns should be more generously proportioned than in land-engine practice.

*Crank shaft.*—The crank shaft especially should have a high factor of safety; in diameter about six-tenths of the cylinder bore; the bearing

surface should be ample, but not exaggerated. It is obvious that a short, stiff crank shaft withstands torsional stress best. Augmenting too much the length of a bearing does not improve it, for it then bears principally at the ends, and the bushes show abnormal wear at their extremities. A safe rule with two-stroke engines is to make the journals as short as the cylinders allow.

*Columns.*—Some firms have already adopted through bolts extending to the cylinder covers, relieving cylinder and columns of all stresses. This is certainly a good feature to ensure lightness and strength, but not stiffness. We may adopt it in some future design for the sake of lightness where weight limitations might be imposed. Such a feature has often been employed and is not the exclusive property of any firm. But considerations of stiffness and stability necessitate dimensions of the columns being such as to guarantee sufficient strength and render through bolts superfluous.

*Bed plate.*—It is needless to say that the bed plate should also be of standard marine type and very strong, and is best supported also in the middle.

A second group comprises those parts that are subjected not only to various stresses caused by piston force, but subjected to high temperatures as well.

Here the problem becomes much more complex. Not only must the designer seek very direct transmission of forces, but he must also provide for expansion in a much larger measure than in the case of a steam engine. He must design castings of uniform sections, allowing uniform expansion; he must avoid also too heavy sections, specially in the neighborhood of the combustion chamber. Here weight is no more synonymous with strength. He is at a loss how to calculate resulting stresses because the initial conditions of his castings are unknown to him. The problem becomes more a question of metallurgy than design, and it is only by close collaboration between the foundry and the designing office and a good deal of experimenting that the right solution will be found.

We call to mind here Professor Junker's study on heat transmission through cylinder walls, and the fact that about 260,000 calories per square meter per hour are transmitted through the walls of the combustion chamber during the combustion period, whereas the heat transmission through the wall of a steam boiler is only 30,000 calories per square meter per hour.

Such enormous quantities of heat . . nsmitted causes very high stresses in the casting, and only special sorts of cast iron can withstand such stresses. It is highly to be desired that more extensive study be given to the question of appropriate cast irons for internal-combustion engines. On the other hand, such intricate castings as a cylinder cover should be suitably heat-treated to relieve them of any casting stresses.

All the parts composing this second group should be very accurately machined, necessary clearances very exactly determined, and relative degree of wear closely followed. Knowledge of the very interesting question of cast iron growth under recurring high temperatures is here all-important. You will agree with me that for this class of pieces a firm with long and varied experience in Diesel engine design is best placed to succeed.* It would be fool-hardiness to attempt to solve such a complex

*For engine builders to attribute all engine failures to personnel is wrong, not only to the personnel, but they are placing themselves in the position of being infallible, which position is false. To state that the figures and calculations show ample strength is an elegant defense to the lay mind, but is not convincing to the engineer.

problem as they present by sheer reasoning and calculations.

*Valve gear and reversing gear.*—The third group comprises all pieces that are subjected neither to high stress nor high temperatures.

They require principally a close study of all phenomena that occur dur-

ing the starting and the maneuvering of the engine. They must be so interlocked as to exclude any error on the part of the engineer in charge, be so designed that too high pressures in the cylinders or high stresses in the valve gear be impossible; must allow the rapid cutting out of any one of the cylinders when necessary; or slow speeds either by uniform reduction of fuel to each cylinder or cutting out of cylinders or groups of cylinders. The fuel must be automatically cut off in case of "racing" or in case of shafting or propeller failure. Every possibility of failure or getting loose must be excluded.

From these considerations it follows that the use of mechanisms operated by compressed air or oil pressure should be confined to those functions where a leak would be of no consequence or where servomotor action is necessary to cope with considerable forces, and an auxiliary mechanism, performing by hand the same functions, should always be provided as a safeguard.

We are strong advocates of simple machines, but this simplicity must not be purely superficial and obtained by the addition of a multiplicity of functions hidden to the view, and we are anxious to know what the average engineer would do when placed before one of these enclosed and mysterious devices that have in some cases contrived under the name reversing gear. In fact, few mechanisms have of late been patented under such various forms as the reversing gear, and yet only a minimum are based on thorough knowledge either of the Diesel engine or marine requirements. Stated briefly, simplicity is here most desirable, while every extra function involves several extra pieces, therefore augmenting the possibilities of failure;* but it must not be bought at the expense of maneuvering qualities, even in the smallest detail. Here, again, collaboration between the marine engineer and the Diesel man is essential.

*In this regard it is believed that, in the interests of simplicity, we ought to abandon reversible engines in submarines. Without this feature greater simplicity will result, as a great number of valves will be done away with. To reverse, and to assure reversing, the motors are generally used to start the engines in the reverse direction. This takes a great deal of power. It is seldom that submarine vessels are run astern except for a short period of time, and the motors are generally designed to stand any momentary overload which an emergency would require.

*Accessibility.*—Having firmly established that reliability is the feature of first importance, we would say that the engine should be as accessible as the first feature permits. We can nowhere find better models than in the marine steam engine, which forty years of experience at sea has perfected; and it is therefore quite natural to adhere closely to its standards wherever possible. Many firms have certainly made the great mistake of not studying the marine steam engine closely before they decided on what seemed to them the right design. If they lacked designers with marine-engine practice they would have done well to let some of their force serve a few months of apprenticeship on some tramp steamer. It would certainly have gone far towards avoiding certain features in some designs that are rather extraordinary from the marine engineer's point of view.

Besides, there is another big advantage in adhering closely to marine standards. As it is utterly impossible to give marine engineers a wide knowledge of the Diesel engine in a short time, and as no other than the steam-engine man is available, it is well to confide to his hands only a limited number of functions with which he is not familiar. It is evident that he will have more confidence in an engine of which a majority of features are quite familiar to him than in an entirely new one of perhaps the enclosed type. He must be able to *see* how every piece is performing its function and to *hear* it as well.

Besides, it is a mistake to believe that the ideal engine must be entirely automatic in all its functions. It is good to keep the men on watch wide awake by giving them something to do and obliging them to make frequent rounds. In the German navy, for instance, it is considered important that an engine should have certain polished parts, that must be fre-

quently cleaned and kept bright. In so doing the engineer becomes familiar with his engine and often detects small defects that might have developed into serious failures.

We therefore advocate the open type, with visible bearings, crossheads, &c.; also easily to be touched without having some door to open and going through the dangerous process of groping in the dark.

*The piston.*—The piston is a most essential organ. It requires to be designed properly and accurately cast of suitable material. In the usual type, the trunk piston, it must fulfil double requirements. It takes the side pressure of the connecting rod and must also be gastight. Opinions vary as to possibility of its successfully meeting with both requirements.

We have reason to know, and are backed in this assertion by thirteen years' experience, that this double problem offers no very great difficulty in all medium-sized engines; but cases of gripping have nevertheless occurred from time to time. It is, of course, safer in a marine engine to exclude all possibilities of such accidents, and their eventual results, by providing cross-heads, thereby leaving to the piston, or rather the piston rings, the one function of making a tight joint. Ample clearances can then be provided that exclude all such accidents as mentioned above. Besides, the piston pin is placed in much more favorable conditions of work, and its temperature can be more easily controlled; the crosshead guides are made adjustable and can be efficiently cooled.

We are pleased to recognize that this sound principle has been universally followed. It has the one disadvantage of being considerably more expensive than the other design, but the augmented safety it brings cannot be bought at too high a price.

Suitable means must also be provided for cooling pistons of large diameter. Four-stroke land engines are known to us where pistons of 600 mm. diameter and about 5 m. piston speed have been left uncooled and are giving satisfaction; in fact, it is a matter of suitable material and design as to where the demarcation line must be drawn. But, true to our principle of taking no unnecessary risks, we consider that pistons should be cooled from about 150 horsepower per cylinder upwards.

In large marine engines with an open bed plate, absolute tightness of piston cooling joints is not as essential as in, for instance, a submarine engine of the enclosed type, where sea water mixed with the oil has disastrous corrosive action on the bearings. Any suitable device may be used, the simpler the better, provided it is reasonably tight and there is no objection to the use of sea water.

### DISMANTLING.

In developing a design it must also be kept in mind that all pieces requiring frequent or even occasional inspection must be easy to remove without involving a large amount of dismantling. In this first period of development it will be well to make more frequent revisions than is really necessary for safety's sake; the piston, the bearings, the valves must be made as easily accessible as possible.

The question of the best method of removing a piston has been much debated. We consider that all designs, even of small engines, necessitating the removal of the cylinder to get at the piston should be excluded, unless those gears that are generally supported in brackets fixed to the cylinder be supported entirely by the columns below the cylinders, in such a way that the cylinder can be removed without necessitating the dismantling of these gears. Such a device, however, makes the use of long push rods necessary. These considerably augment the masses to be accelerated at each valve opening and necessitate very strong valve springs, for which space is not always available should their dimensions exclude high fatigue and consequent lameness.

Another method consists in removing the piston from below. It is

attractive from several points of view, and we have no objection to it for certain types of engines, but in some existing designs it has made a rather undesirable construction of the cylinder liner necessary, of which we cannot entirely approve. It also entails a higher machine, therefore a heavier one.

A third method consists in taking off the cylinder cover and removing the piston from above. It is certainly the simplest and most direct method, provided it does not make the removal of the cam shaft necessary. It is true that it necessitates breaking high-pressure joints, but this is quite a secondary consideration when suitable joint material is provided that can be used over and over again. We do not advocate this method for all types indiscriminately; available head space is the decisive factor for its employment. Besides, here is a point where actual experience of both systems should necessarily decide.

### VALVES.

An organ that necessitates very frequent revisions is the fuel valve. It is therefore highly important not only that the needle should be removable in a minimum of time, but the valve seating and the pulverizer as well, and this removal should not make resetting of the valve gear afterwards necessary. Towards preventing the necessity of all too frequent revision we would recommend such designs as give the needle a minimum of chance to stick. We also recommend that consideration be given to the influence of expansion of the needle and valve gear on the timing of the valve. The starting air valve has also given trouble; it must under no circumstances be so designed that it can rust in its casing.

This is an important point, as sticking fast of the starting valve has given rise occasionally to serious trouble, even accidents; but these troubles can be avoided also by right choice of materials and clearances. We think it would be a good feature to provide some means of grinding the valve on its seat from time to time while the engine is running, perhaps to go as far as to provide positive closing of the valve through some other means than spring action.

In general, the whole valve arrangement to be made inadequate requires long experience with Diesels; neither do we consider that any of the present arrangements are ideal ones.

The valve gear should be as silent as possible, but silence should not be obtained by too far enclosing it.

### FUEL PUMP.

Fuel pumps are often capricious, refusing service for no apparent reason, necessitating regrinding of the valves, &c. Here also the right design and very accurate machining are essential.

It would carry us too far to establish a complete characteristic of the fuel pump. Let us summarize the principal features. Air pockets should be absolutely avoided. Valves should not be given too broad a seat; the fuel must be very finely filtered, the fuel pipe from the pump to the fuel valve must be as short as possible. In marine engines, as the possible reduction of the number of revolutions of the engine at slow speed is largely dependent on a very equal amount of oil being given to all cylinders, we would recommend a separate plunger for each cylinder and not a fuel distributor. Here also suction and discharge valves should be easily removed and replaced.

### COMPRESSOR.

This is a most important chapter common to both two-stroke and four-stroke engines. We hear rather frequently of failure of this most essen-

tial part. Accidents fatal to human life have even happened and have not a little contributed in shaking public confidence in Diesel engines.

It is evident that at first view the necessity of compressing and storing air at nearly 1,000 pounds per square inch pressure makes a strong argument against the Diesel engine; and yet we affirm very positively that a compressor when correctly designed and necessary precautions taken, brings with it far less danger than, for instance, a steam-boiler plant and steam pipes.

The great fault made by most firms is in oiling their compressors too profusely, or in allowing too much oil to enter the compressor cylinders, and then in providing very insufficient or no means of separating that oil after each stage of compression; also in not sufficiently intercooling the air. The air must be cooled immediately after the discharge valve of each stage. Long uncooled pipe strains between compressor and cooler should always be avoided and the receivers should be of large size. It must not be forgotten in calculating a compressor that the air must reach in each stage a pressure considerably higher than the receiver pressure in measure to open the valves, and the number of stages must be so chosen that the final temperatures be never high enough to cause ignition of the oil vapors carried in the air.

On the other hand, safety valves should be placed immediately after the discharge valves and not at the coolers or receivers, because an explosion propagating in a long pipe produces a local rise in pressure in that pipe sufficient to blow it up long before the explosion wave reaches a safety valve placed further on.

The air bottles should be provided with sufficient drainage conveniently located; in fact, the best method is to provide a continuous blow off. This does not entail a large loss of air and contributes highly towards safety.

On the other hand, perfecting the pulverizer should also tend towards lessening the necessary injection pressure. There are very numerous compressor-valve designs more or less suitable. We will not discuss them and their relative merits; we would only say that a suitably designed valve and valve seating of suitable materials have a very long life, and may have no tendency whatever to stick as long as they are not oiled with too much profusion.

The air bottles, when tested, all at double their working pressure, offer every guarantee of safety and have only failed in those cases where subjected to oil-vapor explosions. The same remark applies to air pipes. Besides, an air-pipe explosion is far less dangerous to human life than the explosion of a steam pipe. A leak is without consequence and without danger, and can be easily repaired. The same does always apply, to our knowledge, to steam pipes. The necessary high-pressure air pipes are all of small dimensions.

In fact, we maintain that a conveniently designed air generating set can be made absolutely reliable.

### FOUR-STROKE OR TWO-STROKE.

We have now discussed a certain number of principles of design that apply to all types, but we have not decided the question of system. Is our marine engine to be a four-stroke or a two-stroke?

For powers up to about 1,500 horsepower we do not deny that the four-stroke can be applied about as well as the two-stroke, and that their application up to those powers has been attended by considerable success. There is a good reason pleaded in favor of the application of that system. As equivalent sizes of cylinders to those applied existed already in land-engine practice and had given but little difficulty, and as the four-stroke engine is much easier to build than a two-stroke, for a firm untrained in Diesel engine practice, it is natural that it should have been chosen.

If, however, the two-stroke is an engine that cannot be created by other than Diesel specialists, there are other important considerations to be expounded, when deciding the cycle to be chosen. It is to be expected now that the marine Diesel has more or less passed the experimental stage, that higher powers than 2,000 will be in demand. A well-established policy should seek to foresee all eventualities.

It is established that above certain powers the two-stroke for obvious reasons comes alone into consideration. The four-stroke builder cannot indefinitely augment the number of cylinders of his engines or base them on steadily augmenting mean effective pressures. The time must come when the strongest advocates of the four-stroke must necessarily turn to the two-stroke for further expansion. But, as we mentioned above, two-strokes are not created in a few months time; before tackling the marine two-stroke a firm needs sufficient experience in land engines of the same type, and it is natural to conclude, as the two-stroke is necessarily the system of the future, that it is a better policy to build no intermediary type of another system, but grapple at once with the essential problem.

Four-stroke builders will claim that the two-stroke is as heavy and as expensive as the four-stroke. It may or may not be true. Impartial comparison is at present impossible because the four-stroke builder has already taken advantage of all possibilities of reducing the weight of his engines, while the two-stroke builder has essentially had other factors in mind.

Computing the mean effective pressure at which the best-known four-stroke marine engines work, it appears to be very high; in fact, considerably above the average considered as sound practice in land installations. This means that these engines at their nominal power are already overloaded, and that it would be very unwise to seek further augmentation of the specific power on these lines.

On the contrary, the best-known two-strokes now in service use a very low mean effective pressure. This is the natural result of the maker's desire to remain on the safe side and to build engines that can smoothly develop the power for which they were sold. It means also that the two-stroke builder for obvious reasons has wittingly not taken advantage of every possibility of his system during the first period of development. But there is no doubt whatever that this mean effective pressure can be considerably augmented, and that the day is very near when the two-stroke will give more than double the power of the corresponding four-stroke of same cylinder volume.

Assuming that the two-stroke to be so designed that the weight of the charge of air imprisoned in the cylinder be considerably augmented and that the pulverizer be so perfected that it assures perfect pulverization of an augmented quantity of fuel, we obtain a corresponding rise in effective pressure. Such means cannot be employed in the four-stroke without the adjunction of auxiliary pumps. Even if this step were taken in a four-stroke, it involves danger to the exhaust valves. Higher indicated pressure means higher exhaust pressure and exhaust temperature. This is without consequence in a two-stroke; the bars between the exhaust ports are water-cooled, and the whole cooling water of the cylinder being compelled to pass these bars, their cooling is very efficient. Further, they are also cooled by the excess scavenging air after exhaust.

Not so the exhaust valves. Although an exhaust valve can be water cooled, it is subject to very high temperatures during combustion and is not cooled by the incoming air as efficiently as a scavenging valve or suction valve. Besides high temperatures, it must withstand a constant hammering action, and as it is necessarily heavy when water cooled, small wonder that the most frequent trouble with four-stroke comes from this source.

A second objection is that the fuel consumption of a four-stroke is con-

siderably lower. We will not contest this point; we remark only that this difference in fuel consumption must not be exaggerated. It is actually about 10 per cent. to 15 per cent. at the most, and we incline to think that the difference in first cost of the engines, when prices have reached their right level, will ultimately more than compensate this difference to the advantage of the two-stroke.

Admitting a two-stroke mean effective pressure to be double that of the four-stroke, the same power can be obtained with half the number of cylinders of the same dimension. This means a much shorter engine, a very appreciable advantage were it only considered from the point of effective balance and stiffness, and not of cost and weight. It will have besides the same degree of uniformity of turning moment.

It might be objected that a large number of cylinders is favorable to good balance, and that for the same number of cylinders and uniform crank angles assumed, the four-stroke is better balanced. We think that it is not advisable to go too far in balancing requirements, because balance must not be only theoretically perfect but effective, and a very long engine cannot be made stiff enough to be considered, as a whole, absolutely rigid around its center of gravity. Besides, experience has proved that the engine can be theoretically balanced and yet give rise to strong vibration according to its position in the ship and due to unequal torque. We think that a very even torque is the essential factor and we are reminded of Fram's studies on the problem of degree of balance and sort of balance necessary according to the engine's position.

Four-cylinder two-stroke marine engines have given no trouble when made stiff enough, as actual experience on board ship has proved. Four cylinders is the minimum number possible in a two-stroke, if it should start from any position of the crank shaft, whereas this number is six in a four-stroke if the best crank angles for balance have been chosen, and unless the starting valve opens during 180 degrees. Even then the first impulse would be insignificant and a very high exhaust pressure involved, with the consequent conditions it would impose in the design of the valve gear.

The scavenging pump of a two-stroke is an organ that can be made absolutely reliable. We hear little or nothing of failures to this part of the machinery and its dimensions can be augmented very largely without entailing the use of high-grade cast iron or necessitating special workmanship. It is a source of far less trouble than, for instance, the air pump of a steam engine. It can be made very light while subjected to no high pressures, and therefore does not involve great expense and is certainly cheaper than the extra cylinders or larger dimensions the four-stroke involves.

As to the question of drive, we consider that rocking levers off the connecting rods are best. Driving a scavenge pump directly from the crank shaft gives a nice looking design; but as soon as the necessity appears of putting two engines, one behind the other, on one line of shafting, it involves a scavenge-pump crank as heavy as the combustion-cylinder cranks, as it has to transmit the whole torque of the engine.

We will not attempt to decide if automatic valves or a mechanically-driven piston valve is best. We consider they are equally good, perhaps the second is slightly more on the safe side. For the sake of safety also it is well to provide two scavenging pumps, allowing the ship to continue on reduced speed should one of the pumps fail.

Since the accident on the Continent to a big 6,000 horsepower engine the opinion prevails that scavenging pumps are dangerous. We need only remark that this accident was more a fire than an explosion; that it was occasioned by the ignition of oil vapors in the low-pressure receiver. Moreover, the vapors ought never to have been there had the necessary precautions been taken towards preventing the scavenge air carrying too

much oil with it. Besides, a very unsuitable sort of lubricating oil was employed. Let it be remarked here that oil vapors should not only be prevented in compressors and scavenge pumps, but also in the enclosed crank cases of both the two and four-stroke high-speed type. When forced lubrication is employed the oil should be kept at a definite maximum pressure by a spring-loaded overflow or any suitable device; the bearings should have but the strictly necessary side clearance, and a profusion of oil grooves should be avoided.

In fact, most accidents which have happened to the two-stroke were through no fault inherent to the system, but were due to unsuitable design, conceived with abnormal haste and insufficient practical knowledge of the whole problem. Many people still consider the whole two-stroke problem as unsolved. They see in the two-stroke a capricious and mysterious device, that will for no apparent reason occasionally refuse to start, or show apparently unexplainable variations in specific power according to the number of cylinders, &c. Yet with sound and exhaustive tests as a basis, a slow-speed two-stroke can be designed for a given power with as much security as the four-stroke.

To conclude, we would call to mind that the reversing gear of a two-stroke is considerably simpler than that of its rival.

### SINGLE-ACTING OR DOUBLE-ACTING.

We abstain from discussing fully the double-acting engine, for we maintain that although it may be the type of the future for very high powers, single-acting engines have such very marked advantages for all powers up to perhaps 5,000 that the double-acting engine hardly comes into consideration at present.

Besides, the double-acting engine is infinitely more difficult to build, and we feel that there is advantage in acquiring a few more years' experience with the other type before we attack it. Otherwise one would be in some measure in the same position as a student placed before a system of equations with more unknown factors than are compatible with a definite solution. Therefore let us for the present be more modest in our aims and seek satisfaction in attempting to bring the single-acting type to perfection.

### SHALL OUR TWO-STROKE ENGINE BE A VALVELESS ENGINE OR PROVIDED WITH VALVES? SHALL THESE VALVES BE LOCATED IN THE CYLINDER HEAD OR AT THE EXHAUST BELT?

As we have already explained, there is no doubt that the best scavenging, therefore the highest efficiency, is given by the direct method of scavenging from above. The specific power of a valveless engine is decidedly too low to allow of its use as a very high-powered engine, and its overload capacity is limited by the fact that the scavenge ports are necessarily closed before the exhaust ports.

Valves located at the exhaust belt are advantageous in many ways, but there can be no doubt that with this system also scavenging is not as perfect. True, this engine can be overloaded, but it is an open question whether this actually does more than compensate for the imperfect scavenging, unless very high compression pressures be provided for.

We do not pretend that our present design of two-stroke is final, that we shall apply it to all powers. Every progressive firm always has future developments in study, but we would emphasize that in all our experience of two-strokes, the scavenge valves have given us no trouble whatever. We must bear in mind that they are efficiently cooled by the incoming air.

It may be objected that scavenge valves in the cylinder cover involve a complicated casting. We do not attempt to conceal that we had trouble in the very beginning with cylinder covers where cast steel was used.

We have since found a most suitable material for these covers and are very satisfied with the results. In fact, we see no necessity to alter our design in any way for those powers contemplated in the near future.

### WHAT ARE THE MOST SUITABLE MEANS OF WORKING THE AUXILIARIES?

Naval experts know what large quantities of steam auxiliaries can absorb under circumstances, and it is most important to decide how they are to be worked in a Diesel-engined ship.

It seems to us far too expensive to work all of these with compressed air, although we have in the exhaust gases means of heating this air and, to a certain measure, counterbalancing compression heat losses; we nevertheless think this method is not suitable and its cost prohibitive. It would mean either very considerably augmenting the capacity of the main compressor or else providing a large separate engine-driven auxiliary set exclusively for this purpose. Another combination might be to separate the main compressors from their respective engines and combine all to a single group sufficient for all requirements.

Some people advocate the use of the exhaust gases for raising steam. It must not be forgotten that one-third of the heat contained in the fuel is lost in these exhaust gases. Setting roughly the heat consumption of the engine at 2,000 calories, it means a loss of about 600 calories, of which perhaps 50 per cent. might be available—sufficient heat to raise about a pound of steam per horsepower of the engine. This is probably enough to cope with the auxiliaries when at sea, but it would still leave us helpless when in port.

It is possible that the high-powered Diesel of the future will have a large auxiliary group driven by a powerful oil engine, and generating not only electricity and compressed air for all purposes, fuel injection included, but the scavenge air as well and driving also all auxiliary water pumps. By such means a notable augmentation of the power per shaft could be obtained. This system has already been employed, we think, in the most recent four-stroke marine installations. Besides, it would then be possible by varying the speed of the auxiliary to adapt the quantities of injection and scavenging air generated to the load in a more economical and rational manner than at present.

It is to be objected that this would scarcely contribute towards a quiet engine room unless the group were in an enclosed separate space, and it is all-important that the engineers' attention be not distracted from the main engine. Whatever it may be, the question of auxiliaries drive is a complex one than can only be decided by comparative results of several systems.

For the present we would advocate the old system and drive the auxiliaries by steam up to perhaps the whistle and the steering gear, which might be driven by compressed air, but only when at sea. It is, for the present, the safest system, and has the advantage of entrusting to engineers' hands a type of machinery which they are used to handle.

### CONCLUSION.

We have already pointed out that close collaboration between shipbuilding firms, possessing long experience with marine steam engines, and Diesel firms is necessary—even essential—to ensure success. We mean collaboration in the best sense of the word, not the half-hearted help of people prejudiced for no essential reason against the new type of prime mover. It is this collaboration we have endeavored to find.

We appreciate the value of the help we have found in this country, but we recognize also that our German collaborators have shown themselves particularly progressive, energetic and efficient. We quite understand

that English marine circles are conservative; that this is dictated by prudence is a high quality, and we praise this conservatism. We have even pointed out that continental builders have sometimes shown lack of this essential quality. But prudence must not be blind to evidence, and we think that there is now no reason for the distrust shown by the English public in the oil engine.

We in no way wish to deny that there are still unsolved problems in the Diesel marine engine, but for ships of the commercial type these problems are certainly of minor importance. To be impartial one must not forget the difficulties encountered with marine steam engines at the outset or with steam turbines. Our conviction that the marine Diesel engine has a big future before it is absolute, and is based on an already long experience.

We sincerely hope that progress will not be seriously impeded by lack of enterprise, half-heartedness of designing staffs, or the ill-will of the sailing engineer.

Firms that have made every sacrifice to further the cause of the Diesel and are in a large measure responsible for its present stage of perfection, cannot indefinitely progress alone, but must be met half way in their endeavor. We trust that prospects will soon justify a more extensive manufacture of the marine type of Diesels, and give consequent possibility to make it more and more, not only from the engineering point of view, but financially a successful rival of the steam engine.

The following discussion of Mr. Carels' paper is taken from "Shipbuilding and Shipping Record":

DISCUSSION.

Mr. J. T. Milton, Chief Engineer Surveyor of Lloyd's Register, said he had followed the progress of Diesel marine engines through the world so far as they had gone, and had taken great pains to keep himself up to date from the point of view of their reliability at sea—a point of view far more important to those dealing with the safety of ships than the point of view of economy. He was much indebted to the makers of Diesel engines for the very frank way in which they had discussed their troubles with him whenever he had seen them, Mr. Carels as well as the others. If the makers would only confer freely with one another, as he was pleased to say they had with him, the troubles of the Diesel engine would be very much minimized, and they would have Diesel engines very much more reliable than at present. If the suspicion of unreliability were taken away there would be a great many more Diesel engines made. Instead of losing their business by communicating with their colleagues, makers would gain by it, and he was quite sure there would be such a demand for Diesel engines that the present makers could not cope with the inquiries. He gave some of his experiences regarding the reliability of Diesel engines at sea. The first Diesel engine they had had dealings with was the *Vulcanus*, made at Amsterdam by the Works Board. The engine was a four-stroke cycle engine. She had had some troubles, but they had been very few. At Lloyd's they kept a list of every one of the Diesel-engined ships. They watched them run as they went on their journeys and they kept a record. From that record he found that the *Vulcanus* had been a remarkably successful ship. Her owners—and it was a very wise thing to do—would not send her far from home at first. They engaged her in short runs. Then they had one little trouble with her. On one occasion she came into port with her pistons seized through the oil not being burned in the cylinders. Until the particular voyage, she had been using a very light oil, but on that voyage she was supplied with one of the oils from the East, not a distilled oil.

### PISTON TROUBLES.

When her pistons seized, they had put it down to the oil. Then they found it was not the oil; and now she was every day burning the same oil, without any change being made in her construction. The trouble had been that the exhaust pipes were lined with fire bricks, and the fire bricks got misplaced and choked, but they did not know that at the time. Now she had been sent out to the East, to a place where there was no workshop for doing repairs, but the owners had much confidence in her. The *Selandia* began to run in February of last year. On her first voyage she had a little trouble with two of her pistons and with some of her exhaust valves. The difficulty with her pistons arose from want of lubrication, but if, on a ship with sixteen pistons, only two gave trouble and the others had given no trouble ever since, they could not say that the design of her pistons was wrong. On her arrival home her exhaust valves were altered, and she had since had no trouble at all. These two vessels were classed with Lloyd's.

All he knew of the *Jutlandia*, which was similarly engined, was from the shipping papers, as she was not classed with Lloyd's. Once fairly started, she had been exceptionally favored until just the last few days, when she was in a collision, a circumstance, of course, not due to the engines. The *Christian X* put into Queenstown with her fuel valves deranged. The cause of this was that she had been on a voyage to South America and had there got some oil which was more combustible than the oil used in their general experience, and had ignited a little bit quicker than the others. When she got into exceedingly heavy weather and they had to ease the engines down as slow as ever she would run, the valve timing for the ignition of the oil gave a little pre-ignition, and that was the cause of the trouble.

### TWO-STROKE AND FOUR-STROKE.

In the further course of his paper the author says the large slow-speed two-stroke gas engine had not been a big success; but for obvious reasons the Diesel two-stroke did not have to contend with the same difficulties as the gas engine. Its fuel consumption was relatively low, its reliability and capacity for overload could be made greater than that of the four-stroke, and the mean effective pressure achieved showed how it could become a dangerous rival of the four-stroke for high powers.

At the same period a high-speed four-stroke engine of 400 H.P. at 300 revolutions was produced; it was tested by Prof. Eberle, the results being published in several German periodicals. It did much towards establishing the conviction that the combustion can be equally good at high speeds, and that the mechanical efficiency of a four-stroke engine varies only slightly from low to high speeds, sufficient valve area and right timing provided. Its indicator diagrams showed also that the constant-pressure combustion was not the best one for all fuels and all speeds, and that there was an advantage in so timing the injection of fuel that a slight explosion occurred immediately after the dead center. The idea that the Diesel was necessarily a slow-combustion engine was done away with. Here again a new and vast field appeared to open for the application of the Diesel engine.

The 100-H.P. reversible two-stroke engine exhibited at Milan in 1907 proved that reversing a Diesel was not only to be accomplished with relative simple means, but that this reversing could be obtained in as short or shorter a time than with a steam engine.

Designers with experience in one type of prime mover exclusively are apt to be a little one-sided; being used to certain standard shapes and methods they lack initiative to try new features, and are sometimes unjustly prejudiced against certain designs and their applicability.

### CYLINDER TESTS.

As soon as Diesel patents became public property many very capable men with wide experience in gas and steam engines worked out very interesting and independent designs and did not consider it necessary to stick so closely to original shapes of cylinders and valve gear. We have no objection to inventive ability when it is backed by sufficient experience; and this ability has produced amongst other things many beautiful designs. Some of these, especially in high-powered slow-speed engines, have been good and have notably furthered the Diesel cause.

### CYLINDER VALVES.

As to cylinder valves, they could only read in Mr. Carels' paper that there had been trouble with cylinders. If people would only show their designs and criticise them, one with another, the speaker thought the difficulties of cylinder covers would be soon over. As to difficulties with air compressors, air compressing was not a new subject at all. It was used in Her Majesty's Navy 30 years ago, and certainly 25 years ago there were compressors made to compress air up to 1,000 pounds. They were compressing at the present time for service purposes up to 2,500 pounds. Air compression ought to be the most simple mechanical problem they had to face, but there had been more trouble with air compressing in connection with Diesel engines than with any other part of the engine.

At the very high air pressures used in Diesel engines—and compression means high temperatures—oil did not behave as one expected it to behave. Most oil, even the very high-grade oils, in some way carbonized at high pressure and at high temperatures. The carbonization of the oil in the compressors was one thing that gave a great deal of trouble. Lubricating oil absolutely got decomposed in the form of coke. While he was on his last visit to the Continent he visited several of the engine works and discussed the matter of air compressors, particularly with the Diesel-engine makers. They were very free in their communications to him. He found that they did not properly realize the position. He found that, in several instances, the pipes of the air compressors had burst. In one case they put down the burst to the pipes being made of steel. At another works he found that they had always had copper pipes, but had had several bursts in between the high-pressure compressor and the cooler. In another works he was told they had bursts between the low-pressure and the intermediate stage, and between the intermediate stage and the high. These bursts of air pipes were due to the combustion of the oil inside the pipes.

### THE QUESTION OF AUXILIARIES.

He had been asked what his views were as to the working of the necessary auxiliaries of the Diesel engine. He thought that for any ordinary marine engine of reasonable power, it was far and away better to work the air compressors and the scavenging pumps from the main engine and not from any auxiliary, but in large vessels it was advisable to let the main engine drive the screw, and only the screw, and to have an auxiliary to drive the compressor and the scavenger pump. He did not think that Mr. Carels was right in saying that the two-stroke cycle engine was necessarily 10 per cent. less efficient than the four-stroke engine. It did not take that 10 per cent., if they had a reasonably good design, to drive their scavenging pump. In the four-stroke engine, they had two idle strokes and the engine friction going on all the time. What they wanted was not the fuel capacity for indicated horsepower, but the fuel capacity for brake horsepower.

### FUEL CAPACITY.

With regard to fuel capacity, the very thing which tended to economy tended to durability in all these engines. He had found that the oil was not completely burned until the engine stroke in some four-stroke and two-stroke engines. He had found that the temperature at the end of the stroke was even greater that it was before the end of the stroke. In some of the newer designs that did not happen. They got combustion more complete in the earlier part of the stroke, although there was more heat in the earlier part of the stroke and less at the finish. One of the Burmeister engines had gone out to Bangkok and back again and the exhaust valves were perfect, not damaged at all, since they had a new fuel valve. Mr. Carels did not mention what he made his cylinder covers of. He said "a material," but of course it was cast iron. The words "cast iron" conveyed as much to many engineers as the word "wood" did. They got many kinds of cast iron. He was very much interested, two years ago, to see the way in which, in one of our factories, at least, a very great deal of care was being taken to produce cast iron of high tensile strength and good resistance to impact. He thought if they had to make Diesel engines, they should be very careful with regard to the quality of the cast iron which was used.

### PIONEER WORK AND CONSERVATISM.

Mr. D. B. Morison said his experience of the Diesel engine had been on the commercial side rather than on the technical. In older times, people collected their tears in bottles to show the extent of their grief. The pioneer, from the commercial point of view, of Diesel engines would require a bucket, not a bottle. They were all greatly indebted to Mr. Carels for his masterly review of a prime mover which was ultimately destined to play a leading part in the world's work. Mr. Carels complained of the conservatism of marine engineers, but was it not that conservatism which had built up the steam engine until its reliability was such that the time table of an Atlantic liner was now comparable to that of a mail train?—It was that question of the reliability of the marine steam engine which was responsible for some, at least, of the hesitation to embark on the actual development of marine oil engines. Moreover, that pioneer work was of a highly costly character, and although it was not impossible to find many progressive shipowners, their progressiveness was invariably associated with commercial instincts which were so highly developed that, whenever the baby fell sick, it was the engine builder who always found it in his arms and was the recipient of much generous sympathy on behalf of himself and his offspring. In order to perfect the Diesel marine engine so that it might be successfully managed at sea, there must be a close collaboration between marine engineers and the makers of Diesel engines. Much harm was done about two years ago by the extravagant claims made by irresponsible people, and that had been intensified by the sensational rise in the price of oil. The adoption of marine oil engines, would, in his opinion, be very gradual, but ultimately success was certain.

### TRAINED ENGINEERS.

There was the question of engineering personnel. When trouble had resulted with the oil engine the engineer had no pals with whom he could consult, and he got disheartened and troubled and expense resulted. The engineers of an oil engine required to have a special training. They had to diagnose the various diseases of the engine. When on a trial trip, the speaker was much struck with a small difficulty which occurred, which difficulty was put down to the air compressor. It took about an hour and a half to overhaul the air compressor, and the air

compressor was quite innocent. It was some little fault in the cylinder tubes. It interested him very much, because there were a good many experts on board and no one seemed to be able to put his finger on the trouble. With regard to the *Eavestone,* Mr. Westgarth was closely associated with Messrs. Carels in the design, and everything possible was done by that firm to guide them in the road to success. Messrs. Carels advised them to have—and they adopted—cylinders of cast steel. Only practical experience revealed the fact that cast steel was not suitable for Diesel engines. It developed small hair cracks, nothing of an alarming nature, except that if extended, they would have resulted in a blow up. All these troubles had been overcome, and the only difficulty they had now was with the wearing of the cylinders. Whether that wearing would continue they did not know. They hoped to have a consultation with Mr. Carels to ascertain what was best to be done. It was essential, if the oil engine was to develop, that reliability must be established. It did not matter to the shipowner whether his ships were propelled by oil engines, steam engines or gas engines. He was out to make money, and if he got a boat that gave trouble and at the end of the year his balance sheet was not favorable, he was not likely to repeat whatever the prime mover was. If oil engines were to develop quickly, the makers of them must work together. They had now a splendid opportunity on that coast. He suggested that they should join together and give the Armstrong College a big engine—an engine of a sensible size—not a toy—to be exhaustively tested and subjected to the exhaustive analysis and research of which they all knew Prof. Weighton to be capable. They all knew the value the steam engine at the college had been to them and he was certain that an oil engine would be an equal value.

The following comments on Mr. Carels' paper appeared in "The Engineer."

M. Carels commenced by an admission that the Diesel engine is not making the rapid progress at sea which was expected, and accounts for it partly by reason of the high price of fuel and partly by a sense of disappointment that the lavish promises made by the—he is careful to say continental—technical papers have not been fulfilled. In this statement he excludes the technical Press of this country, and gives it credit for the clear and impartial manner in which it has dealt with the question of the Diesel engine. We are, however, inclined to add other reasons—the high cost of the Diesel engine, the impossibility of obtaining quick delivery at a time when freights are high, and the fact that the majority of shipowners cannot convince themselves that the Diesel engine has as yet actually proved itself in every way suited to their needs. They note that there are a number of motor ships afloat, and naturally they prefer to wait and see how those ships acquit themselves rather than add themselves to the number of pioneers. This is an attitude of which the British nation is sometimes said to be inordinately fond, to its own detriment; but, on the whole, it may be doubted if our nation shows much more caution than others. Indeed, M. Carels himself shows how right the attitude was and, to a certain extent, still is. He points out, for example, the effect which the demand for light-weight high-speed engines for submarine purposes had on design in general—most of the firms who took up the construction of the Diesel engine beginning with this type—and then gives a number of reasons for the unwisdom of the course pursued. He admits, however, that it was valuable in showing how a Diesel engine should not be made, and draws attention to the present policy of most continental builders, which is to return to more moderate speeds and fewer types. Bearing on the same general question, he riddles the belief held in some quarters that any firm can set to work to construct Diesel engines without previous experience, pointing out that

the original Diesel engine owed its success entirely to the exact experimental and laboratory work which preceded its appearance. This is a point on which we ourselves feel somewhat strongly, and we have often pointed out the danger of attempting to put on the market a Diesel engine without very extensive preliminary bench tests.

Another question which shipowners are waiting to see settled—if settled it ever can be—is the superior merit of the two or four-cycle. M. Carels gives his preference to the two-cycle on account of the absence of exhaust valves, the simplicity and cheapness of its reversing gear, and the better turning moment for the same number of cylinders—not for the same power, as he puts it. He admits, however, the higher fuel and lubricating-oil consumption and the higher mean temperature of the two-cycle. Having plumped for the two-cycle engine, he discusses the *pros* and *cons* of scavenging valves and cylinder ports. We have on previous occasions expressed our doubt as to the completeness of the scavenging with the port or valveless type of engine, but we have never been able to obtain sufficient figures upon which to base a definite opinion. M. Carels states definitely that the port type of engine must give a lower mean effective pressure and higher fuel consumption, but in view of the fact that the Sulzer engine of the *Monte Penedo*—which, by the way, had a combination of ports and valves—has a consumption of only 0.46 pound per brake horsepower per hour, we do not think there can be very much in the point. The simplicity of the port design, as evidenced by the Diesels Motorer A.B. engine, must, however, be admitted, though it is probably true that there is a limit to the revolution speed of this type. Actually, M. Carels deems the port type most suitable where simplicity and maximum reliability are the essential factors, but he contends that in cases where fuel economy is the factor, the four-cycle engine should be used up to certain powers, while for high powers—say, from 1,000 horsepower upwards—the two-cycle scavenge-valve engine best meets the case. Probably it is not safe to go further than this at the present time, and we must wait the accumulation of reports from ships fitted with engines of both kinds to reach a more definite decision. M. Carels, however, takes up the position that "The marine Diesel engine is no longer an experiment, but has given proof of its quality," and in that we are not disposed wholly to agree with him. The Diesel-engined ships actually at sea have not, as far as our information goes, given as little trouble in the way of attention and renewals as the marine superintendent has become accustomed to expect with his steam engines. Not that the troubles with some of the ships have been serious, but in some cases they have been serious enough to cause a slight feeling of anxiety. Certainly almost every new boat shows some modification from the original design of engine. In this connection, too, there is the factor of the human element to consider, and although it may be perfectly true that the ordinary steam engineer may find but little difficulty in accustom-himself to think and work to the necessary thousandths instead of his thirty-twos, he cannot but find the conditions in a Diesel engine room very different from those to which he has been accustomed. For some time to come the Diesel engine must be and should be regarded as an experiment, or we shall too soon reach standardization and arrest developments. M. Carels is, however, very sound when he points out that whereas the final overall efficiency available to drive a boat is, in the case of a steam engine, only some 7 or 8 per cent., it will amount to some 17 to 22 per cent. in the case of a Diesel engine. This, of course, largely boils itself down into the question of the price of the respective fuels, and it may roughly be taken, from this point of view, that a Diesel engine can compete on an equality with the steam engine when the price of oil is about three times that of coal. But, in marine engineering, economy can never take the place in the slightest degree of reliability. First

and foremost, the engines must be free from breakdowns; afterwards, and only afterwards, can economy be considered. M. Carels is not slow to recognize this obvious fact, and he deals ably with the different conditions under which the various parts of the engine have to work, and gives us some useful hints. He states, for example, that the diameter of the crank shaft should be about six-tenths of the cylinder bore, and draws attention to the fact that the bearings should not be too long, as, unless the shaft is exceedingly stiff, it bears principally at the ends. This would appear to mean that bearings are likely to be on the short side, and a natural deduction is that some form of forced lubrication is a *sine qua non*.

Other points on which the author touches we must leave the reader to find for himself in the paper. In many cases they bring us back to our contention that the Diesel marine engine is still in the experimental stage, and we note with no little pleasure M. Carels' observation that "It would be foolhardiness to attempt to solve such a complex problem by sheer reasoning and calculations." Much must be discovered by experience, and this is best to be obtained by the collaboration of the marine engineer and the Diesel engine builder. The author points out that some firms have made the mistake of not studying marine practice before starting on their designs, and in this connection he re-echoes the views which we have often expressed on this particular aspect of the question. We are glad to note, too, that M. Carels agrees with us in stating that the trunk piston is not a satisfactory job for marine work, even though it is used extensively in land practice; he admits that even there cases of seizure have occurred. But we find ourselves unable to agree with him in his somewhat casual ideas as to the necessary water-tightness of the joints in piston-cooling arrangements. Even in an open engine, where there is not much danger of the escaped water mixing with the oil and causing it to saponify, we think everyone would admit that such leaks can certainly do no good, and, in our opinion, they tend to make the engineer less careful, as he will avoid approaching places where he may get wet, and he will become somewhat hopeless about keeping the appearance of his engine up to a high standard. Perhaps we rather labor what many may consider a small point, but seeing that absolute tightness is obtainable, we cannot subscribe to any slackness in this direction. We were recently on board the *Eavestone*, and noted that the glands of the piston-cooling water pipes had not been packed for three months, and yet were absolutely dry. If this is possible in one case, it must be obtained in all cases. We think M. Carels is wise in insisting on accessibility, especially in view of his remark that "In this first period of development it will be well to make more frequent revisions than is really necessary for safety's sake." In the Carels engine the piston is, of course, removed from above by taking off a cover, and we are informed that it can be taken out for inspection in about an hour. Our own view on the desirability of removing pistons from below are well known. M. Carels is, apparently, not quite decided in his mind as to what are the best means of working the auxiliaries, but he suggests that in high-power driven engines in future they will all be driven by an independent oil engine of considerable power, but is very wise in raising the objection that this may make a noisier engine room, though that is a difficulty which we hope to see overcome.

We have done no more than allude to a few of the points with which M. Carels dealt, but we shall have done good service if we induce any reader to turn to the paper itself. It is unquestionably the most practical paper on the marine Diesel engine which has yet appeared, and an example is set to other makers which we shall be glad to see them follow. In too many cases makers, when they do write papers on their own products, are concerned more with praise of what they do themselves and with

oblique dispraise of what others do, than with the broad issues which are of interest to all. We are persuaded that the more openly such issues are discussed, the more does the industry as a whole benefit, and we rejoice, therefore, over a paper of this kind.

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## DIESEL MOTORS IN SUBMARINES.

This comment is good as it shows the foreign tendencies in submarine engineering. They seem to be reverting to a type that we are getting away from, and the reasons are very good.

An article which appeared in "Le Yacht," at the end of last year, dealing with Diesel motors on French submarines, has not been allowed to remain entirely unchallenged. On the appearance of the Diesel motor such firms as Harlé, Schneider, La Loire, Normand and Delaunay-Belleville took up the construction of the new engine, especially of the larger type, for application to submarines. A series of 16 submarines of 400 tons was started in 1911 with the *Brumaire*. This type has two 420-H.P. 6-cylinder Diesel motors, instead of four cylinders, as is the case in the *Emeraude* series. Their weight is 17 tons, and they make 400 r.p.m. at a speed of 13½ knots. Though the above-mentioned horsepower is nothing exceptional, it is only with the greatest difficulty that it could be obtained; in fact, there is not one boat of this series which had not to undergo protracted trials. The greatest and most frequent trouble seems to have been cracks in the cylinder and piston. The gases in the cylinder, according to the article in "Le Yacht," reach a very high temperature during the working of the motor, viz: about 1,100 to 1,200 degrees at the highest point of compression with a simultaneous pressure of 35 to 40 kg. The outer casing is kept at about 400 degrees by water or oil cooling. Further, the tops of the pistons become white hot, whilst the heat in the interior is scarcely red hot. The expansion is, consequently, very uneven, and acts detrimentally on the metal. The air compressor is another difficulty. It is worked by means of an auxiliary shaft set in action by the main shaft, and has to compress air in a bottle up to 75 kg. pressure, which expands to 35 or 40 kg. in the cylinders. The slightest trouble in the regulation of the compressor hinders the pressure from rising, and thus restricts the horsepower, which remains far from what it ought to have been. Further, the slightest introduction of lubricating oil into the compression chamber causes the metal to fracture.

Dealing with the alleged causes of these difficulties, Herr Manfred Erhardt, in an article in our contemporary "Schiffbau," argues that they are based on technical errors. According to him, the French engines scarcely allow temperatures of 400 degrees in the outer casing of the Diesel engine, and even if 400 degrees were to be taken as absolute temperatures, they would still be too high. The cylinder temperatures in high-speed Diesel motors amount, as a rule, to about 40 or 50 degrees C., and the exhaust pipes, with good water-cooling arrangement, to about the same. The temperature of the crank case is, of course, less. Higher temperatures could only occur if non-cooled scavenging-pump jackets and non-cooled high-pressure water pipes are used. But in this case also, theoretically speaking, it would scarcely be beyond 63 degrees C. The high-pressure pipes of the individual compressors having a temperature of about 200 to 300 degrees are strongly insulated and do not radiate. A temperature such as mentioned by the author of the article would make the atmosphere in the engine room, to say the least, extremely uncomfortable. In this respect the high-speed Diesel engine is not different from the stationary Diesel motor, which has proved efficient for years.

The pistons never become white hot outside or red hot inside. The facts are as follows: Owing to the high number of revolutions and the consequent rapid succession of ignitions, the four-cycle and the two-cycle engines require the interior cooling of the piston. More commonly a so-called "fire plate" of Siemens-Martin steel of conical shape is fitted in the middle, this being the portion of the piston mostly affected by the temperature of combustion. In engines built after the type of the submarine Diesel motor, these fire plates, after the engines had worked continuously for six days with full load, showed colors which for this material belong to temperatures ranging between 600 and 700 degrees C., but not absolutely.

On the other side of the piston the temperature is scarcely higher than that of the cooling agent, viz: 60 to 70 degrees C. for oil, and 90 to 95 degrees for water. Moreover, a simple calculation of the expansion of a piston passing from a cold condition to the red-hot or even white-hot condition shows at once the impossibility of the above-mentioned condition, leaving all other phenomena quite out of consideration.

The working of the compressor by means of an auxiliary shaft seems altogether an unhappy idea, which cannot be justified either on the ground of space or other questions of construction. There is certainly not a single one of the more important German constructions with this device. (This may be questioned.)

As to the fractures to the metal attributed to lubricating oil having forced its way into the compressor, no doubt ignition, *i. e.*, explosion of the oil destined for lubricating purposes, is meant. The fact, according to Herr Ehrhardt, has been of exceedingly rare occurrence in his experience, and the possibility of it must be reduced to its proper proportion. The piston compressors, as is well known, must be divided into such gradations or stages that the pressure, *i.e.*, the temperature variations in the individual stages, be not too great. For instance, an injection pressure of 75 atmospheres requires a compressor of three stages. In that case there would be in the first stage about 4.3 atmospheres absorbed; in the second about 18.4 atmospheres absorbed; and in the third stage about 76 atmospheres absorbed. The condensation temperature brought after each condensation down to the initial temperature amounts adiabatically to 180 degrees C. to 455 degrees absorbed. The lubricating oil to be used for the high-speed Diesel possesses a flash point which lies above this temperature. Moreover, a local ignition of a very small amount of lubricating oil is harmless if otherwise lubricating is carried out carefully. A collection of lubricating oil, however, can be avoided by removing all obnoxious spaces.

Whatever may be said on these and other difficulties in connection with the Diesel motor for submarines, even French experience—which Herr Ehrhardt does not consider as equal to that of some other countries—has proved that these motors are both capable of standing long runs as well as capable of further development in power for the above purpose. According to the article in "Le Yacht" above referred to, the authorities have decided to pass from 420 H.P. in the *Brumaire*, to 550 H.P. in the *Clorinde* and the *Cornélie* which have the same displacement, but 15 knots in speed. In the *Marionette* the power is increased to 720 H.P. This craft is designed for from 530 to 630 tons. Both motors are four-cycle and of the 6-cylinder type, each of 120 H.P.—"Shipbuilding and Shipping Record."

## NAVAL AND MILITARY AVIATION.

During the past year all the great Powers have continued to devote increasing attention to the aeroplane as an instrument of warfare. Indeed, it is becoming more and more clearly recognized by all that, as we

have repeatedly urged, the employment of the craft by the navies and armies of the world is likely to be the main, if not the sole, source of orders for the industry.

The aeroplane armed with a quick-firing gun is now almost a commonplace and, given but little time, we should see it developed sufficiently to make a most effective reply to any offensive airship, which, surviving the hundred and one chances of destruction by natural forces, would dare to sail over an enemy's territory. Although all the leading makers of aeroplanes have probably turned their attention to this matter, the mounting of a quick-firing gun on machines turned out by Vickers, Short Brothers, the Grahame-White Company, Henry Farman, and the Blériot firm may be specially mentioned. Experiment and invention have proceeded, too, in the matter of dropping bombs from aircraft, although the experience gained on this point in Tripoli and the Balkans does not, to say the least of it, offer much encouragement. The Italians, a recent writer says, found it a perfectly futile proceeding, even from the moderate altitudes at which the incompetence of the enemy permitted them to practice it. The bombs very frequently failed to explode, and when they did, did little damage, because it was found "inconvenient" to carry high explosives in the aeroplanes. Further, since an aeroplane cannot hover over a given spot it was found all but impossible to take aim before discharging the projectile. In the Balkan war it is reported thirty bombs were dropped in one day on Adrianople from a Bulgarian machine. These succeeded in killing or wounding only six people.

It may indeed well be doubted, as we pointed out nearly two years ago, whether armed aircraft will ever be used to attack troops, although no doubt the moral effect of their presence would be considerable. We are confident that the prime value of the aeroplane in war always will be the facility it affords for the rapid collection and delivery of information concerning the enemy's movements, and that the main object to be sought in arming them either with quick-firing guns or with bomb-dropping devices is the power to checkmate the enemy's scouts similarly mounted. Experienced officers at the army maneuvers affirm that the air scout promised to cause a radical change in tactics, as it is no longer possible to move troops save under the cover of mist or darkness. Something, no doubt, might be done in the way of attacking and blowing up the opposite party's magazines, and it may be noted that the new naval magazines at Portsmouth have been specially designed to resist this form of attack. They consist of a number of ferro-concrete chambers sunk into the earth, covered with about 4 feet of soil and turfed over so as to make them as nearly as possible invisible from above.

The demand for the armed aeroplane has given rise to a vast amount of invention, and it seems as if the immediate evolution of the flying machine is going to be almost entirely regulated by it. Apart from the design and construction of guns especially intended for such use—a matter not quite within the sphere of this article—we have already seen the beginning of the influence which the demand will have. For instance, the fact that all monoplanes and many biplanes are now equipped with tractor screws makes it difficult to fire a gun in the direction of flight, yet such a proceeding is obviously desirable in the case of one aeroplane pursuing another and when overtaking a hostile dirigible baloon. It has been proposed to overcome the difficulty in several ways, none of which is free from objection. In one case the gun and its operator may be mounted below and to the side of the pilot. With this arrangement it is scarcely possible to fire straight ahead unless the gun is displaced very considerably from the center line of the machine, and if this is done the whole design of the aeroplane is profoundly modified and rendered at variance with accepted ideas by the lowness of the center of gravity. A second alternative is to mount the tractor screw on a hollow shaft and

fire through this. Obviously, however, the angle through which the gun can be trained with such an arrangement is very restricted. A third method which has been proposed is to link up the trigger of the gun with the crank shaft in such a way as to lock it twice or oftener in each revolution, namely, just at those instants when a propeller blade is passing across the mouth of the gun barrel. With this we are clearly more or less unrestricted as to where the gun may be placed, but we have grave doubts as to whether or not the draught of air from the screw would not very materially influence the accuracy of the shooting. Further, the mechanism required to effect the locking of the trigger would clearly have to be made automatically adjustable for movements of the gun in the horizontal plane, and would probably be complex and very liable to get out of order.

There is one other alternative in the case of monoplanes, and this, we think, shows clearly the great influence which the arming of the aeroplane is going to have on its design. We refer to the abandonment of the now general tractor screw and the reversion to the old-fashioned propeller. Such a plan seems to get over many of the difficulties, even although it introduces certain aeronautical disadvantages and losses. At the Paris Show, December, 1913, a Borel monoplane was exhibited in which the screw was at the rear, and in which a small Hotchkiss gun was carried at the front end.

The arming of aeroplane naturally suggests the armoring of it as a reply, and it is interesting to note that at least one firm—the Blériot—has actually constructed a machine in which the body is protected from rifle fire by means of steel plates. Here, again, the influence of the gun will, we believe, have a marked effect on the evolution of the aeroplane, for it is reasonable to hope that the armoring of the machine will lead ultimately to the universal adoption of the "all-steel" type of construction. Clearly, there is very little object in protecting the bodies of the pilot and passenger from rifle fire so long as a lucky shot is able to splinter and smash some vital part of the machine, and send its occupants to death in a different way. In process of time we are even prepared to see the same influence at work in leading to the abandonment of canvas as a covering for the wings. At the present moment any aeroplane would be quickly rendered non-effective by directing the fire of an automatic gun for a few moments on to the wide mark afforded by its sustaining surfaces. It would be a mere waste of time to neglect this means of destruction in favor of attempts on the pilot's life.

### FURTHER DEVELOPMENTS.

Apart from these essentially military developments, progress has been active in many other directions during the year. But, wherever we turn, we find that in the minds of those responsible for the developments in question the military use of the aeroplane is never very far in the background, and that so far but little progress has been made towards establishing the craft as a means of commercial transport.

Thus, in the attempts which have been made—without much success, it must be said—to silence the aeronautical engine, it is the purposes of war which are being served. At present the approach of an aeroplane, even at a considerable height, is foretold by the unmistakable hum of its motor, and to the civilian it is not desirable, in the present state of the art, that it should be otherwise. In warfare, however, it would go a long way to discount the value of the aeroplane as a means of scouting, for even already troops are being trained to take cover from aircraft so as to hide their existence from the observer or to mislead him as to their numbers, movements and nature. For instance, troops marching on an open road are not allowed to occupy the full width of it, but are confined to

one side in the hope that if an aeroplane passes overhead the observer will mistake the formation for a narrow part of the highway. A train of artillery is made to simulate a mounted detachment of engineers, and *vice versa* and so on. The silent aeroplane engine is, however, not yet universal, and so long as the radial rotary type maintains its present popularity it seems doubtful if it ever will be.

Next to making the aeroplane silent the best thing to do is to make it invisible, and in this direction distinct progress has to be recorded. By means of a new "dope," the wing fabric can now be rendered practically transparent, so that at anything but a low altitude the machine under most circumstances should be all but invisible. We have not seen a machine thus treated in flight, but we fancy that at certain angles in bright sunshine there would be a strong reflection of light from the wings which might, instead of making its presence invisible, make it all the more visible.

It may be questioned, however, whether a stronger construction of wing than can be secured by the use of fabric, treated or untreated, is not more urgent from the military point of view than the attainment of invisibility. Signs are not wanting that progress in this direction will soon have to be recorded. It is not altogether for the sake of lightness that fabric is used so universally for the wing covering. The fact that it results in a flexibility sufficient for warping purposes has also a large part in the reasons for its adoption. Get rid of the necessity for warping—which at best is an undesirable proceeding mechanically—and quite half the reason for using fabric becomes invalid. It is therefore of some significance that during the past year a tendency to revert to ailerons as a means of checking the rolling of an aeroplane has become manifest among several of the leading makers.

It is perhaps too early yet to hope that we shall soon see the wings covered with light sheet metal, although even now we are glad to note that a serious beginning is being made, as in the Clement-Bayard and R.E.P. monoplanes and in the biplanes of the German Aircraft Works of Leipzig, with an "all-steel" construction for the framework of the wings and chassis. Even wood, we believe, would be better than fabric as a wing covering, for it would not be nearly so susceptible to total destruction on being punctured, although if not treated in some way it might be troublesome by reason of its warping and its power of absorbing moisture. In this connection, it may be mentioned that from America comes news of the invention of reinforced wood—soft wood, that is, strengthened by strips of sheet iron embedded in it and coated electro-metallically on its surface. The metallic coating, it is said, acts not only as a protection against moisture, but renders the wood fireproof and indicates very readily the existence of strain at any point.

### AEROPLANE ENGINES.

Turning to the plant required for propelling the aeroplane, we note, with something like dismay, the ever-increasing demand for higher and still higher-powered engines. Two or three years ago 50 to 75-horsepower was thought sufficient. Today machines equipped with motors, such as the twenty-cylinder, 200-horsepower Anzani, are not uncommon. In so far as this increase means an augmented reserve of power, it is all to the good from the military point of view, for here questions of running expense are of very secondary importance, and a wide range between the highest and lowest speed of the machine is everything. But from all other standpoints than that of warfare, this trend towards higher powers only confirms us in our opinion that the aeroplane, for some time to come, at least, will remain a military, and not become a commercial, means of transport.

While it is true that the Gnome radial rotary engine and its numerous imitators still hold the field for popularity, it is also true that the stationary water-cooled motor, the aerial development of the motor-car engine, is steadily gaining in favor. Nor is the reason far to seek. The rotary motor is extremely wasteful of oil and fuel, while the other type is by comparison highly efficient. Indeed, with the length of flights now attempted—such as that made during November by Captain Longcroft and a passenger, who flew with but one stop from Montrose to Portsmouth and then to Farnborough, a total distance of 630 miles, in 7¼ hours—the weight saved by the use of a rotary engine is practically discounted by the weight of the extra oil and fuel required.

That the ordinary type of engine is more trustworthy as well as more efficient than the radial rotary air-cooled type, seems, too, to be the lesson of the aeroplane motor competition held in Germany towards the close of 1912. Of the seven air-cooled engines entered, only one succeeded in passing the main seven hours' test, and did so only on its second attempt; while of the thirty-six water-cooled engines, seventeen passed without trouble. It will be interesting to see whether this verdict is supported by the results of our own Naval and Military Aeroplane Motor Competition to be held in April.

Into the progress which has been made in the design and construction of aeroplane engines, we cannot go at any length. Developments have, however, been extensively made, both in the details of the motors and their general design. The general trend is still towards the production of a light-weight motor occupying a small space and developing a high power with a good efficiency. In view of the fact that machines capable of lifting several passengers are now being regularly constructed and flown, it is somewhat surprising to note that the demand for lightness in the engine plant is still as keen as, if not keener than, ever. Yet we are positively assured that such is the case. It would appear, in fact, that aeroplane builders, while thinking nothing of adding pounds to the weight of their machines in the shape of luxurious upholstering for the pilot's and passenger's seat, and similar items, insist upon every possible ounce being saved in the weight of the motor.

### AIRSHIPS.

Disastrous as the history of the airship has been in the past, all records are exceeded by the catastrophes of 1913. Conspicuous among the calamities of the past twelve months are the destruction of the Zeppelin Ersatz Z 1 at Karlsruhe in March, of the German army's Schutte-Lanz at Schneidemühl in July, of the German navy's Zeppelin L 1 off Heligoland in September, and of the latter's companion, the L 2, at Johannisthal in October.

In spite, however, of these unfortunate additions to its already more than unfortunate record, the dirigible balloon, we regret to say, is regarded more favorably than ever by the authorities in this and other countries. Germany, Austria and Italy, Britain, France and Russia are all undertaking the formation of an extensive airship fleet, although we cannot see that the dirigible balloon of today is constructionally in the least superior to its forerunner of, say, five years ago.

In this country airship activity has in especial suffered a regrettable revival during the year. Statistics on the subject are not available officially, but we believe that we now possess six airships in active commission, that at least five are under construction—four at Vickers' Barrow works, and one, a rigid, at Armstrong, Whitworth's works at Newcastle—and that negotiations are being conducted for the acquisition of others of foreign construction. Of those in actual use, two—the Astra Torres and the Parseval—have been purchased during the year. It is probable that

others of the same type are contemplated, while the Italian Forlanini airships are known to have attracted the attention of the Admiralty. It may here be noted that in future our Army is to confine its aeronautical activity to aeroplanes, and that the Navy is to take over the airships at present in existence and under construction.

Some little time ago we dealt with the commercial prospects of aeronautics, and in discussing the future of aircraft, expressed the opinion that the running charges were so high that, short of some wholly unforeseen and unlikely development, the aerial transport of passengers or goods could never be made to pay. From some particulars recently published concerning the work of the Zeppelin passenger airship Victoria-Louise, we are now able to illustrate our contention by a reference to actual figures. Between March 4th, 1912, and November 26th, 1913, this vessel made four hundred trips, covering all told a distance of 29,413 miles and occupying in the aggregate 852 hours. During this time she carried 8,551 passengers, and used 18,550 gallons of petrol, 15,276,000 cubic feet of hydrogen and 2,123 gallons of lubricating oil. Taking the cost of petrol at 1s. 6d. per gallon, of hydrogen at 10s. per 1,000 cubic feet —this figure has been supplied to us by the British Oxygen Company, and does not include any cost for carriage, a very heavy item by itself— and of oil at 4s. 6d. per gallon, we find that the running charges during the time stated amounted to:—Petrol, £1,391; hydrogen, £7,638; oil, £472; total, £9,501. The cost of fuel, oil and gas thus amounted to £11 3s per hour of flight. The average speed was about 35 miles an hour, and the average number of passengers carried about 21. Hence, per passenger mile these three items of the running charges represented an expenditure of over 3½d. Add to this the wages of the crew, the interest on the capital and the depreciation of the airship and its sheds, and one can well believe that aerial journeys of this nature cannot hope to rival other modes of transport on a financial basis.—"The Engineer."

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## THE PERISCOPE FOR SUBMARINES.

### The German Institute of Naval Architects.

The second paper dealt with on Thursday, November 20th, discussed "The Development of the Submarine Boat Periscope," and was read by Dr. Weidert, of Friedenau-Berlin. In the course of a historical introduction it was set forth by the author that a submarine-boat sight tube had been invented in 1854 by Marié Davy. About fifty years later such tubes had begun to take practical shape. The author traced the development of the appliance from the vertical tube with inclined mirrors at top and bottom, on the lower one of which an image reflected on the upper one could be seen through various arrangements of prisms and lenses, to the perigraph—afterwards called "periscope"—of the French Colonel Mangin. This was essentially a ring-shaped mirror, which enabled photographic panoramic pictures to be made right round the horizon. When this was applied to the top of a sight tube of a submarine the picture was reflected downwards on to a mirror within the vessel. Inasmuch as the ring pictures were small and distorted, an arrangement had to be made by which a disc picture of any desired piece of the horizon could be reflected into the space within the ring picture. The round view of the horizon was obtained partly at the expense of the disc picture, which was reduced in size by the width of the ring, and it was considered doubtful whether this arrangement would maintain its ground.

Binocular sight tubes, which were subsequently tried, necessitated either

an increase of the diameter of the tube or a reduction in the size of the picture. A field-glass arrangement was then fixed at the bottom of the tube, the observer, as before, looking, horizontally at the prism there fixed.

To determine the magnetic azimuth at which an object appeared from the submarine a compass was fitted in the head of the sight tube. An attendant advantage of this was that the compass was in this position least affected by the steel masses of the vessel's structure. An arrangement could also thereby be made such that the observer at the sight tube could at the same time keep the course, steered by the submarine, under observation.

The distance of objects of known dimensions from the submarine was determined by so-called telemeter plates, which were graduated vertically and horizontally in hundredths. If an object under observation measured 5 graduations, its distance was 100 times the corresponding real dimension divided by 5. Several variations of this system were stated by Herr Weidert to be in use or proposed. In the double-picture micrometer arrangement two pictures of the same object cut each other in the lens. The pictures could be shifted with reference to one another till, say, the top of the funnel of a ship under observation in the one picture was level with the water of the other. The amount of shift necessary for this thus served as a scale for the measurement of the distance. An appliance for such measurement was illustrated. The sight tubes ranged up to 23 feet in length and between 3 inches and 6 inches in external diameter.

## AIR SHIPS.

A paper on "Analogies between Airship and Seaship Building," by the late Felix Pietzker, of the German Admiralty, Berlin, was then read by the Secretary. Herr Pietzker, who was trained as a navel architect, was, after some years of successful work at the German Admiralty, transferred to the Airship Department. He was a member of the Committee appointed to take over the naval airship L2, and lost his life in the recent explosion of this vessel. His paper contained a popular description of the different types of airships, and showed where points of contact occur between them and ships that sail the seas. He showed that the airship had most resemblance to a sumbarine boat; the grand difference being that a cubic meter of displacement, which for the airship had a lifting power of from one kilo. to 1.2 kilo., in the submarine was equivalent to 1,000 kilos. In an airship instanced, a lifting power of 22,000 kilos. bore a weight made up of 16,500 kilos. of vessel, machinery and equipment, and 1,100 kilos. of crew, the balance consisting of ballast, fuel and passengers.

During a voyage the airship was subject to great differences of expansion of gas within and of air without, which, however, fairly kept pace with each other. In addition to these came changes of temperature caused by variations in degree of sunshine met with and similar causes, which were sometimes very irregular in their nature. In the Zeppelin airships increases of temperature of this kind, amounting to from 10 degrees to 12 degrees C., were experienced. A certain loss in the gas holders due to expansion and contraction occurred on every voyage. When the vessel above referred to rose to a height of 1,000 m., a loss of 2,200 kilos. of supporting power was experienced.

As regards construction, two types of airship were possible—the one with a tensely inflated envelope, and the other with a rigid framework. Of the former of these, the Parseval and the Siemens-Schuckert ships—illustrations of which were given—were examples. The outer envelopes

of these were kept tense, and indeed the general form was preserved by means of gas pressure which had to be kept at a given degree of intensity by the greater or less inflation of a number of air bags or balloonettes within the structure. Outlet valves were fitted to the air bags and safety valves to the outer envelope. The outer envelope could be inflated till the air was completely driven out of the air bags, and any excess of gas was then enabled to escape through the safety valves. The capacity of the air bags—two in the Parseval and three in the Siemens-Schuckert—represented the greatest possible differences in the expansion of the gas, and therefore determined the height to which the airship could rise.

The Zeppelin airship—also illustrated—was shown to contain a number of different gas cells which might be either tensely or incompletely inflated. The safety valves were at the lower sides of these cells to allow heavy impurities to be the first to escape. The outlet valves were on the top. The Schütte-Lanz airship—also illustrated—had a similar arrangement.

The weight of an airship became less as fuel was used up, but might be suddenly increased by snow and rain by amounts ranging from 1½ tons up to 2 tons. Equilibrium was seldom present, and the vertical course had to be kept by the aid of the side rudders. The difference between the navigation of a submarine boat and that of an airship was shown to consist in the main in the much greater influence of the temperature and in the loss of weight due to consumption of fuel in the latter vessel.

The proportions of airships differed in many respects from those of sea ships. The midship section was circular and a too great proportion of length to diameter had, though in a higher degree, the same disadvantages as in a sea ship. Stability and steering conditions were with slight differences similar to those of a submarine. The resistance of an airship was considered to be unaffected by wave resistance, and to be in many ways analogous to that of a submarine.

As regards forms of least resistance, the author advocated for airships, as for submarines, a full entrance combined with a fine run, and stated that this conclusion was borne out by experiments conducted by Eifel and by Prandtl. With regard to material of construction, he pointed out that a comparatively weak material like wood was superior to a strong material like steel, since individual members made of the latter could not have a thickness small enough for the low tensile strengths required.—"The Engineer."

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## RECENT BRITISH TORPEDO-BOAT DESTROYER CONSTRUCTION.

The 1909–10 order for torpedo craft included no less than twenty boats designed entirely by the Admiralty. Generally speaking, they were relatively shorter and fuller than anything of this type that had yet been laid down, being only 240 feet long and having a normal displacement of 780 tons. The machinery consisted of turbines of very similar power and design to those of the *Beagle* class of the previous year, but instead of having five coal-fired boilers a return was made to oil fuel, the number of boilers being reduced to four. This particular batch of boats now constitutes the Second Destroyer Flotilla and is based on Invergordon. Opinions, as might be expected, naturally differ considerably about this type, though its main defects would appear to be the speed of 27 to 28 knots, which on trial was considerably less than that of contemporary German vessels. The armament is good for the size of ship, and arranged so that interference between guns and torpedo tubes is practically nonexistent. Their short and full hulls render these vessels very apt to be

unnecessarily wet in rough weather, and they plunge somewhat jerkily at times, though on the whole they are good sea boats and good gun platforms. Their shortness has naturally resulted in a somewhat congested fore-and-aft arrangement, with the result that the forward funnel is found in very close proximity to the break of the forecastle and has had to be made nearly double the height of the other two—in order to carry the smoke clear of the bridge—which are themselves of very unequal sizes, with the result of producing a distinctly bizarre and awkward-looking boat. This congested deck, and consequent crowded appearance, was avoided in the twenty ships of the *Acheron* class, which were laid down in 1910–11, by the substitution of three large boilers for the four of the previous class. Two funnels were then made possible, and the general appearance and deck convenience of this type, which is otherwise virtually the same in design, was greatly enhanced. Six of these twenty boats were of a special type, to which we shall now refer, while three others—also specially designed and built by Yarrows—were also included in this year's program.

This construction of nine "special" boats in 1911, after a couple of years' abstention from contractors' own designs, was not by any means an admission that Admiralty designs were not all that they might be, but, while partly a means of gaining experience with types slightly modified from the Government designs, it was mainly a form of cheaply experimenting on details of design of no little importance, but which might have been unsuitable in larger and more important vessels, and which at the same time could not seriously affect the individual value of a few of the units of a big batch in the unlikely event of their not being successful. Thus the *Badger* and *Beaver* were fitted with geared turbines and twin screws whilst the *Acheron* and *Ariel* rejoiced in a different form of hull into which several minor improvements over the standard design were introduced. The *Archer* and *Attack* were fitted with boilers of somewhat special design as well as superheaters. They proved very successful; in fact, instead of the guaranteed speed of 28 knots, the mean speed of the *Archer* for eight hours was 30.3, the power with 94 degrees of superheat being 18,500, compared with the 17,000 obtainable with saturated steam. At cruising speed this influence was equally marked, the vessel obtaining 15 knots when using steam superheated to 1,000 and only 13 knots with saturated steam, the fuel consumption being the same in the two cases. This system of practically trying new developments on such a scale has much to commend it.

Of the "1910–11" class, however, by far the most interesting "special" destroyers were the *Firedrake, Lurcher* and *Oak*. Of rather more displacement than the rest of the class, their length is about 15 feet greater, and consiberably more powerful machinery was installed, although the remaining features—armament, complement, fuel supply, &c.—remained the same. These vessels present many advantages over the standard design. They have clearer and less congested decks, due in part to the adoption of two funnels in place of three, and partly to their increased length; their speed on trial ranged from 32.4 knots in the *Oak*, which was tested at deep load, to 35.3 knots as the average of the eight-hour full-power official trial in the *Lurcher*, which is a phenomenal result to obtain when the other features of the design are taken into account. It is hard to believe that these three vessels do not represent a very material improvement over the two earlier standard Admiralty designs of the period, which appear to involve too full a model below water and too congested a deck arrangement, besides being somewhat on the slow side.

The twenty boats of the 1911–12 type represented a great improvement over the designs of the two preceding years. All these boats except one—H.M.S *Hardy*—are to the uniform Admiralty design. They are 260 feet long by 27 feet 9 inches beam and displace normally about 950 tons. Oil fuel is provided for the four boilers, and twin-screw machinery of 25,000

hosepower has enabled a speed of 32 knots to be obtained. The armament and speed are for once highly satisfactory for boats of this size when the other features of the design are considered. Three 4-inch guns are mounted on the center line, as well as two pairs of 21-inch tubes, making these craft as far as concerns armament and its arrangement—a concomitant influence of almost paramount importance—quite the most powerful craft of their type afloat. So much so, in fact, might this be said to be the case—and that without any exaggeration—that the result of even a day action between three or four of these craft and a single big battleship or battle cruiser need by no means be a foregone conclusion, unless in be in favor of the torpedo craft. For the speed and range of the automobile torpedo are now so great, and the accuracy so improved, as to render shooting at a 600-foot target not merely a blind chance even at 6,000 or 7,000 yards, at which range a 4-inch gun would find a destroyer a difficult mark to hit, but at which a big ship would not be a very difficult target. A 6-inch gun on the big ship would be no better, but the question of the reintroduction of such a weapon, while it has an important influence on destroyer design, is hardly germane to this article. For a solitary 22-knot battleship to be caught by four boats of this type, might prove a very serious, if not a fatal, matter for her, unless her gunlayers could hit destroyers with big guns. Probably the most approved remedy is a 12-inch shrapnel, but the introduction of torpedo craft of this power is tending to force upon the bigger navies a new type of vessel represented for the first time by the British *Aurora*, which is a hybrid between a destroyer and light cruiser, with functions which only a vessel of her dimensions could fulfil. The truth is that the 21-inch hot-air torpedo has outranged the heaviest hand-fed quick-firing gun when the range and relative sizes of target are considered. When once that broad essential fact is realized we shall hear less of the return to secondary batteries for use in battle. True, the 6-inch gun will keep even the biggest destroyer at a distance—that distance being the range at which it can be relied on to hit the destroyer. Outside that range what is the remedy? Apparently, when all is taken into consideration, it is considered better to mount the 6-inch gun in an auxiliary vessel that can stand between the battleship and the destroyer. That this auxiliary vessel, assuming her to be an *Aurora* with six 6-inch guns, will have an unhappy experience at the hands of four "L" class boats is very certain, for the simple reason that, even speed apart, she also is big enough to hit with a torpedo. Many years of experience have shown that the 6-inch gun is the biggest man-handled quick-firer that can reasonably be considered; the weight of its shell—100 pounds—and its muzzle velocity are now practically at their maximum, whereas it is only necessary to add 3 inches to the diameter of the torpedo, and it is on almost level terms as regards range with the 12-inch gun. The intermediate type of ship, whatever may be her *rôle* for foreign stations, seems likely to fall into the same category as the 6-inch gun. Her speed cannot be much improved except at an enormous sacrifice of displacement to machinery weight, whereas the absolute size of the destroyer is now very considerable, and the fighting power for a broadside action considerably greater than that of vessels of much greater tonnage than the two combined.

Reviewing Admiralty destroyer construction in the light of the vessels of the last two years' programs—those now under construction being virtually replicas of last year's boats—it is impossible not to feel satisfied with the type that has been evolved in the last five years. It is hard to see why such a low speed was aimed at for the craft which were built in 1909 and 1910, in view of the average 32 knots of the German boats, except for the fact that they were expected to maintain it at sea; but it must be remembered that the British destroyers attained their 28 knots with oil fuel and are able to maintain it while the fuel lasts, but the German boats

of that period have three coal and one oil-fired boiler, all of which had to be very severely forced on a trial of much less duration in order to obtain their high speed. Although the list of the German vessels built during the period under review has been added for the sake of comparison, it is intended to deal with them separately in a future article. Their contemporary construction is given here to indicate how, year by year, the Admiralty programs have allowed for a very reasonable margin both in size and number. There are now, including the 1913–14 order, no less than 109 British destroyers built and building since April, 1909, compared with the 60—one of which has since been sunk, unfortunately with considerable loss of life—built in Germany. It will be noticed that Schichau, Krupp and the Vulkan Company are the only builders with experience of destroyer construction in Germany, whereas in this country we have no less than twelve and even fourteen firms which have constructed modern torpedo craft. This position can only give us satisfaction.

As regards future construction, it seems most improbable at the moment that any reduction in size or speed should be contemplated, especially the latter, which should always tend to increase. The question, however, of the relative importance of the submarine, which is annually growing in size, speed, and offensive qualities, may cause a diminution to be made in the number of destroyers ordered in succeeding years, though a stop is not likely. Peace maneuvers with submarines are even more misleading than with destroyers, for, whereas in war time a periscope would be anathema and a thing to be deliberately run down, in peace time a weighted broomstick, as was proved during the last maneuvers, was sufficient to cause far greater anxiety to avoid contact with a submerged hull than was ever the case with *vis-à-vis* destroyers attacking with lights out. But without wishing to minimize the excellent work done by the submarines in last year's mimic war, it is impossible to conceive conditions in war time when we could have too many destroyers. Even the altogether overwhelming superiority in boats of this class that the Japanese possessed at Tshushima was not found to be excessive, though the fullest use was not made of them at any period of the war. The direction in which we might reasonably look for further developments in type is in the reintroduction of boats as opposed to destroyers. Possibly something of the *Lurcher* type and speed with two pairs of tubes might be made to fill a most satisfactory *rôle* supplementary to that of an enlarged "M" class destroyer, which type is increasing too much in size properly to fill the *rôle* of torpedo boats as they used to be understood. Nowadays the torpedo can be used to fight and does not depend upon surprise for success.

We hope to deal with foreign construction in a succeeding article.—"The Engineer."

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## THE DEVELOPMENT OF THE TORPEDO-BOAT DESTROYER.

By W. Lambert, Naval Architect.

*Published in the "Shipbuilder."*

The "torpedo-boat destroyer" as a type superseded the old class of "torpedo-boat catchers," or, as they were later called, "torpedo gunboats." These boats were able, seaworthy craft quite capable, by reason of their superior armament, of fulfilling their chief purpose of disabling or sinking the torpedo boats of their day; but as the speed of the latter increased it became necessary to radically modify the chief characteristics of the torpedo gunboat, and the new type thus evolved, being generally similar but more speedy and powerful than torpedo boats, became known as

torpedo-boat destroyers. The scope of the modern destroyer is, however, much wider than that of the earlier type. The destroyer has in the first place absorbed the functions of the torpedo boat, but, unlike the latter, is not operated from local bases only. Destroyer divisions are now attached to and work in conjunction with fleets, and their duties include the important task of scouting, for which destroyers are peculiarly adapted by reason of their mobility, shallow draught, and small degree of visibility.

It is not practicable within the scope of this article to adequately survey the features of destroyer design and practice from the beginning, and as a convenient starting point the British "river" class has been chosen. The vessels of this class constructed by different builders are not exactly duplicates, but the leading particulars are generally as given in Column 1, Table I; while Fig. 1, which illustrates the *Welland*, built by Messrs. Yarrow & Co., shows their general appearance.

The ships of this class are eminently safe, strong, and seaworthy. They were the first destroyers to be built with a raised forecastle, and are in consequence well able to be out in the worst of weather, providing also superior accommodation for their officers and crew. The standard of strength was considerably increased in comparison with the preceding 30-knot class, but this gain was dearly paid for. The speed was reduced from 30 to 25½ knots, though the H.P. was greater by 1,500, while the displacement was nearly doubled. It should, however, in fairness be added that the load on trial was made more commensurate with service conditions, being greatly in excess of the load for the 30-knot boats. The large reduction in the vital factor of speed associated with the river class soon raised the question as to whether the scantlings were not excessive, especially as high-tensile steel was largely employed for the main structural parts, and in later practice the scantlings have been modified. Of the thirty-four boats built to this class, one only, the *Eden*, was fitted with turbine engines, being the fourth destroyer to have such machinery installed. The satisfactory nature of the British Admiralty's experience with turbines at this stage led to the adoption of this type of machinery for all subsequent vessels, which policy, in conjunction with the almost simultaneous introduction of oil fuel, revolutionized the propulsive aspect of the destroyer problem, and marked the most important stage in the development of the present-day vessel.

The two classes of boats which followed the river type, though both installed with turbine machinery and using oil fuel, are in other respects widely different. The one is known as the "Coastal" destroyer class and the other as the "Tribal" class of ocean-going destroyers. The Coastal destroyers are in reality improved torpedo boats, as will be seen from the particulars of this type given in Column 2, Table I. The average speed of the thirty-six boats built to this class exceeded 27 knots, while the radius of action averaged about 1,300 miles at 12 knots, which is quite sufficient for boats intended to cruise from local bases.

Of the Tribal class of ocean-going destroyers twelve vessels were built, the leading particulars being given in Column 3, Table I. The maximum limit of longitudinal stress adopted in these vessels was 8 tons per square inch in tension, and 6½ tons per square inch in compression. These destroyers as a whole achieved excellent results on trial, the average speed as a class being over 34 knots. One of them, the *Tartar* (Fig. 2), broke all records for speed, attaining an average of 35.6 knots over six runs on the measured mile, and 35.3 knots average during a six-hours' trial, although it should not be overlooked that this result was to some extent due to the vessel being tried on the shallow Maplin mile.

In the same year that witnessed the inauguration of the Tribal class, tenders were invited for the construction of a 36-knot destroyer, the intention being to lay one down as an experiment. This boat, the *Swift* (Fig. 3), was eventually built by Messrs. Cammell, Laird & Co., Birken-

TABLE I.

| | 1<br>River class. | 2<br>Coast class. | 3<br>Tribal class. | 4<br>*Beagle* class. | 5<br>*Acorn* class. | 6<br>*Archer* class. | 7<br>1911–12 program. | 8<br>Argentine *Jujuy*. | 9<br>Chilian *Almirante Lynch*. | 10<br>U. S. 1911 program. |
|---|---|---|---|---|---|---|---|---|---|---|
| Length on W.L., feet and inches... | 225 0 | 175 0 | 270 0 | 270 0 | 246 0 | 240 0 | 260 0 | 286 6 | 320 0 | 300 0 |
| Breadth, ft. and ins. | 23 6 | 17 6 | 26 0 | 28 0 | 25 3 | 25 7 | 27 0 | 27 0 | 32 6 | 30 3 |
| Trial draught, feet and inches..... | 9 6 | 5 8 | 8 3 | 9 0 | 7 11 | 8 0 | 9 4 | 8 8½ | 9 10 | 9 3 |
| Trial displacement, tons..... | 550 | 245 | 830 | 900 | 780 | 750 | 935 | 995 | 1,560 | 1,010 |
| No. of screws..... | 2 | 3 | 3 | 3 | 3 | 2 | 2 | 2 | 3 | 2 |
| Machinery..... | Reciprocating. | Parsons turbines. | Parsons turbines. | Parsons turbines. | Parsons turbines. | Brown-Curtis turbines. | Parsons or Bn.–Curtis turbines. | Germania turbines. | Parsons turbines. | Parsons turbines. |
| I.H.P. or S.H.P..... | 7,500 | 4,000 | 14,000 | 12,500 | 13,500 | 16,000 | 24,500 | 24,000 | 29,000 | 16,000 |
| Fuel..... | Coal..... | Oil..... | Oil..... | Coal..... | Oil..... | Oil..... | Oil..... | Coal and oil. | Coal and oil. | Oil. |
| Contract speed on 6-hr. trial, knots. | 25.5 | 26 | 33 | 27 | 27 | 28 | 32 | 32 | 31 | 29 |
| Armament..... | 1 12-pdr...<br>5 6-pdr.<br>2 18-in. TT. | 2 12-pdr...<br>3 18-in. TT. | 3 12-pdr...<br>2 18-in. TT. | 1 4-in.Q.F.<br>3 12-pdr.<br>2 21-inch T.T. | 2 4-in.Q.F.<br>2 12-pdr.<br>2 21-inch T.T. | 2 4-in.Q.F.<br>2 12-pdr.<br>2 21-inch T.T. | 3 4-in. Q.F.<br>2 21-inch T.T. | 4 4-in.Q.F.<br>4 21-inch T.T. | 6 4-in.Q.F.<br>4 Maxims<br>3 18-inch T.T. | 5 4-in.Q.F.<br>4 18-inch T.T. |

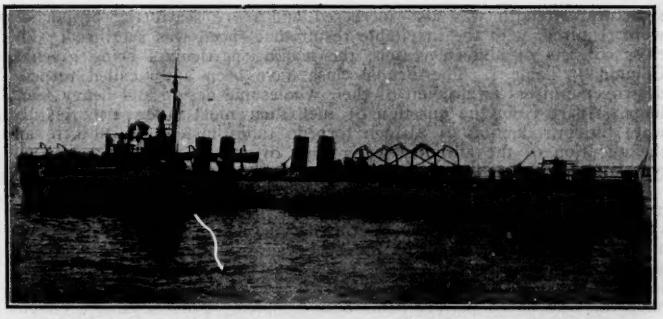

FIG. 1.—H. M. TORPEDO-BOAT DESTROYER "WELLAND," RIVER CLASS.

FIG. 2.—H. M. TORPEDO-BOAT DESTROYER "TARTAR," TRIBAL CLASS.

head, in 1908. She is 345 feet in length, by 34 feet 2 inches beam, and 2,170 tons displacement, and is installed with turbines of 30,000 to 35,000 S.H.P. on four shafts. She carries 180 tons of oil fuel, and her armament consists of four 4-inch guns and two 18-inch torpedo tubes. Although she has the distinction of being the largest destroyer in the world, her general superiority appears to have been hardly worth the extra cost, and this experiment has not been repeated.

The advantages conferred by the use of oil fuel were demonstrated in an inverse way by the introduction of the *Beagle* class of destroyers in 1908–9. Influenced no doubt by the consideration that destroyers would be required for service on stations where oil fuel was not readily obtainable, the Admiralty re-introduced coal as fuel for this class. The

structural strength was also increased and more seaworthy proportions were required, with the inevitable result that speed was sacrificed. This class consists of sixteen vessels, the leading particulars being given in Column 4, Table I. The *Beagle* class have been aptly called torpedo-destroyer cruisers on account of their wholesome design and heavy scantlings. Apart from the question of fuel, their most noteworthy features were the introduction of 21-inch torpedo tubes instead of 18-inch, and the adoption of the all-under-water type of rudder. The new and old types of rudder are illustrated in Figs. 4 and 5.

The 1909–10 British Naval Program witnessed a noteworthy departure in Admiralty practice. Previously a free hand had been given to the various contractors in preparing the designs to fulfil specified requirements, but in this Program all the boats were built to Admiralty design, and in later Programs the same policy has been pursued with the majority of the boats. In the case of only a few vessels with special features have the designs of private firms been adopted. This departure enabled a number of firms to undertake the construction of destroyers who had not previously done so, and had the further advantage of leading to standardization of hull and equipment—at least so far as one particular class was concerned. Twenty destroyers were ordered under the 1909–10 Program, and were known as the *Acorn* class, having the leading particulars given in Column 5, Table I. The machinery is of the Parsons type except in the case of the *Brisk*, which was installed with Brown-Curtis turbines as an experiment. The speed trials of these boats were very successful. The speed for the class averaged just under 28 knots, while one, the *Ruby*, built by Messrs. J. Samuel White & Co., East Cowes, attained 29.3 knots. One of this class, the *Hope*, the first destroyer constructed by Messrs. Swan, Hunter & Wigham Richardson, Wallsend, is illustrated by Fig. 6.

The new Naval Program issued, that of 1910–11, included twenty-three destroyers, fourteen of which were to Admiralty design, being to all practical purposes replicas of those of the previous year, while nine were of special design. Of these latter, five were built by Messrs. Yarrow & Co., Scotstoun, in two groups, two by Messrs. John I. Thornycroft & Co., Southampton, and two by the Parsons Marine Steam Turbine Co., Wallsend. The special features in each case were confined to the machinery, a common feature being the adoption of a two-shaft arrangement of turbines instead of three, as formerly. The leading particulars of the first group of the Yarrow boats, the *Archer* and *Attack*, are given in Column 6, Table I. The turbines are of the Brown-Curtis type, and actually developed 18,500 S.H.P. on trial, the speed resulting being 30.3 knots, an excess of 2.3 knots over contract. Messrs. Yarrow's second group of three also achieved remarkable success. Some 15 feet longer, fitted with Parsons turbines, and having a contract speed of 32 knots, one of them, the *Lurcher*, attained an average speed of 35.3 knots per hour on her official eight-hours' trial, and the other two boats did almost as well. The two Thornycroft boats are noteworthy chiefly by reason of their being the first destroyers to be fitted with Parsons combined impulse and reaction turbines. The hulls of the boats contracted for by the Parsons Company were sub-let to Messrs. William Denny & Brothers, Dumbarton. The most important feature of these two vessels was the introduction of mechanical gearing between the turbines and propellers in order to secure better fuel economy, especially at low powers. They are of similar dimensions to the first Yarrow group, and the contract called for 30 knots speed, which was comfortably attained. The fourteen boats of the Admiralty design again did well, the average speed for the class being just over 28 knots, while the *Ferret*, built by Messrs. J. Samuel White & Co., East Cowes, exceeded 30 knots.

In the case of the destroyers ordered under the Naval Programs of

FIG. 6.—H. M. TORPEDO-BOAT DESTROYER "HOPE," "ACORN" CLASS.

1911–12 and 1912–13 a return was made to higher speeds. Under each Program twenty destroyers were given out by contract, the particulars being generally as given in Column 7, Table I. The propelling machinery consists of either Parsons or Brown-Curtis turbines driving twin screws. One of the vessels of the 1911–12 Program, the *Hardy*, built by Messrs. Thornycroft, has aroused considerable interest by reason of the fact that she has internal-combustion engines for use at cruising speeds. Her radius of action is understood to be increased thereby to more than 4,000 miles. When running at any but cruising speeds, the oil engines, which use the same kind of oil as the boilers, are uncoupled, and when the oil engines are in use the turbines are run in *vacuo*. Fig. 7 illustrates the

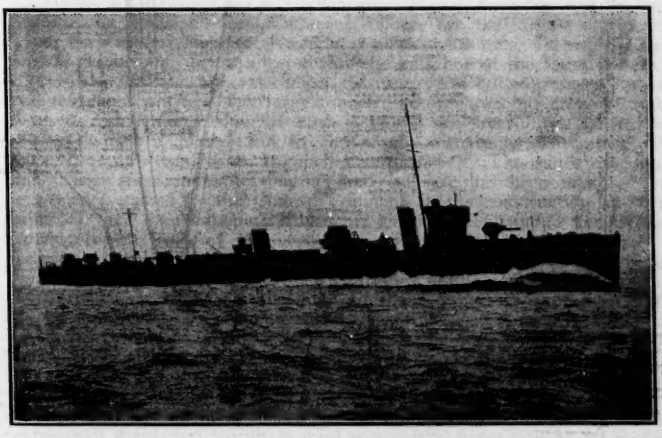

FIG. 7.—H. M. TORPEDO-BOAT DESTROYER "CONTEST," 1911–12 PROGRAM.

*Contest*, of the 1911–12 Program, built by Messrs. R. and W. Hawthorn, Leslie & Co., Hebburn.

Of the sixteen destroyers provided for under the British 1913–14 Program little information is as yet available, but it is understood that the dimensions and speed are similar to the values adopted under the two preceding Programs. The vessels, however, are to have triple screws and geared turbines. Three each have been ordered from Messrs. John Brown & Co. and Yarrow & Co., two each from R. & W. Hawthorn, Leslie & Co., Palmers Shipbuilding & Iron Co., and John I. Thornycroft & Co., and one from Swan, Hunter, and Wigham Richardson, while at the time of writing three more have still to be placed.

The present-day tendencies in destroyer design are undoubtedly in the direction of increased speed, armament and sea-keeping qualities, with a consequent increase in size. In this respect the Chilian destroyers ordered from Messrs. J. Samuel White & Co. are especially noteworthy. Particulars of these vessels are given in Column 9, Table I. One of them, the *Almirante Lynch*, has already undergone trials and attained a mean speed of 31.85 knots on her six-hours' run. Other notable recent vessels are the Argentine destroyers, one of which, the *Jujuy*, was very fully described in No. 28, Vol. VII, of "The Shipbuilder," and the U. S. destroyers authorized under the United States Naval Bill of 1911, which latter are now undergoing trials. Still larger vessels are in course of construction under subsequent United States Naval Bills. The fastest

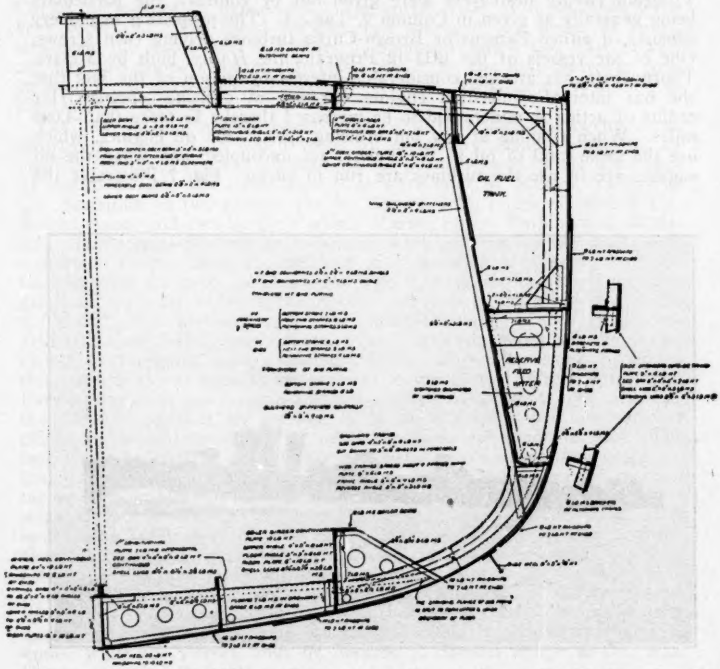


FIG. 12.—MIDSHIP SECTION OF 1,000-TON DESTROYER.

destroyer yet completed is the *Novik*, which has been built for the Russian Navy at the Putilow Works to the designs of the Vulcan-Werke A. G. and attained on her six-hours' trial the remarkable speed of 36.3 knots. This vessel was illustrated in our last issue.

In a typical modern destroyer of 1,000 tons normal displacement the weights are approximately apportioned as follows:

| | Tons. |
|---|---|
| Steel hull | 285 |
| Woodwork | 10 |
| Fittings | 65 |
| Propelling machinery | 482 |
| Armament | 48 |
| Fuel and Stores | 92 |
| Margin | 18 |
| | 1,000 |

In regard to form, some difference in practice exists. The scantlings and structural design suitable for a vessel of this size are indicated in Fig. 12.

Much more could be said on this interesting subject, but in the compass of a short article it is only possible to deal briefly with the leading features, and the author craves the readers' indulgence in regard to any omissions he has made.—"The Shipbuilder."

## PROPULSION OF SHIPS.

### TURBINES OF THE NEW CUNARDER *AQUITANIA*.

Messrs. John Brown & Co., Clydebank, Glasgow, have now got the huge turbines on board the new Cunarder. These are the largest marine engines ever made. Their total weight is about 1,400 tons. In order to permit of the turbines being lowered into the hull, one of the four funnels of the vessel has not been placed in position. Each low-pressure turbine weighs about 425 tons. A large touring motor car could be comfortably housed inside the casing of each. There are over 1,000,000 turbine blades, which if placed end to end would reach over 140 miles. The blades vary in length from 1½ inches to 20 inches. Thousands of pounds have been spent in experimenting, and full use has been made of the experience gained by the Cunard Company in the construction and running of their other turbine ships—the *Carmania*, *Lusitania* and *Mauretania*. In the case of the *Aquitania* an exceptionally high degree of economy is maintained by passing steam successively through high-pressure, intermediate and low-pressure turbines to the condenser instead of, as formerly, direct from the high to the low-pressure turbines. This ensures more work from the steam. Another special feature is the arrangement made for going astern. Each of the four propeller shafts has an independent turbine for this purpose, the two outer shafts being driven by separate high-pressure turbines and the two inner shafts by low-pressure turbines included in the casings of the two low-pressure ahead turbines. This arrangement means the saving of an unusual amount of engine-room space, and permits a most convenient disposition of the starting platform, one of the most important features of the ship.—"The Steamship."

## PROPULSION OF SHIPS.*

By the Hon. Sir Charles Parsons.

The President, in the course of his remarks, said the general upshot of the discussion held in the previous session was that the case for the internal-combustion engine, in its relation to ocean-going tramps, was not so strong as had been supposed, and that in the great majority of instances steam was much the cheaper propellant, having regard to the present prices of fuels and other charges. In the light of a further 12 months' experience there was no doubt that the discussion at that time was a work of great public utility. Last year he dealt with the causes of the difficulties of the large cylinder oil engines, and the case remained as it stood at that time. Dr. Hopkinson's method of internal cooling of the cylinder and valves by injection of water seemed to promise a possible solution, but had not yet been tried on a Diesel engine. Some of the leading makers on the Continent were, however, working on the lines of designing the cylinders, pistons and valves, in fact all parts subject to trouble from excessive heat, in such a way as to be easily and quickly replaced; but this remedy was only a partial one.

The progress of the internal-combustion engine during the past year had been somewhat chequered. The annual report of Lloyd's Register stated that twelve ships were in commission and twenty-five building. He was informed by the secretary of Lloyd's Register that the powers were respectively 22,700 and 43,850, making a total of 66,550 B.H.P. classed under Lloyd's. The report was generally optimistic, but it seemed that the difficulties experienced with the large-cylinder slow-revolution Diesel engine had been greater than the report would suggest.

On the other hand, the adoption of the oil engine for smaller craft had undoubtedly met with great success, and had steadily increased. From data collected by Mr. Edwin Orde, it appeared that, omitting boats of under 100 feet in length, and excluding war vessels, the total number of vessels built and building with internal-combustion engines was 152, with a total B.H.P. of 170,000. The date of the vessels' completion covered the period from 1903 to the present and onwards. It appeared that the average individual B.H.P. was tending to increase, but it should be observed that comparatively few vessels of over 1,000 H.P. had been more than one year in service, and that the majority of the vessels of larger power on order were not yet completed.

Last year he described the three systems of reduction gearing between the turbine and the propeller, the hydraulic transmitter of Dr. Föttinger, electrical gearing and mechanical gearing, and though gearing was not mentioned in the report of Lloyd's Register, rapid and considerable progress had been made during the year. Dr. Föttinger stated that the aggregate B.H.P. of his hydraulic gearing ordered within the last twelve months was about 100,000, including one liner of 20,000 H.P., one war vessel of 30,000 H.P., one cruiser of 45,000 H.P., and one excursion steamer of 6,000 H.P. The last had been delivered.

Electrical gearing had, however, made small progress. The United States collier *Jupiter*, referred to last year, so far as he was aware, had not completed her trials. Messrs. Swan, Hunter & Wigham Richardson had completed a cargo vessel 250 feet in length, the propelling machinery consisting of two Mirrlees Diesel six-cylinder engines, each developing 300 B.H.P. at 400 r.p.m., and driving a Mavor and Coulson three-phase alternator, supplying current to a 500 B.H.P. Mavor and Coulson induction motor, which drove the screw shafting. The vessel, he understood,

*Extracts from the Presidential Address delivered Oct. 31st, before the N. E. C. Inst. of Engineers and Shipbuilders, by Sir Charles Parsons, at the beginning of his second year as President.

had completed her trials satisfactorily. In addition, in America several fire floats of about 800 H.P. had been equipped with electrical gear for working the propeller, fire pumps and other machinery.

Mechanical gearing had considerably extended in use during the year, and the horsepower on order or at work now exceeded 400,000 B.H.P. The vessels to which it was applied were of almost every class. In some cases the whole of the power was transmitted; in others the main turbines were directly connected, and the cruising turbines only were geared to the main shafting. It seemed probable that the use of geared turbines would continue to extend for all classes of ships, and that by its use considerable further economies would gradually be effected, more especially in conjunction with high degrees of superheat. Up to the present, no case of disablement of gear or indeed any material wear of the teeth had, he believed, occurred in any case.

Though superheating was not mentioned in the report of Lloyd's Register, yet it had quickly made great strides, and was now employed in the mercantile marine to the extent of upwards of 1,000,000 horsepower on the Schmidt system alone; nearly the whole was in conjunction with reciprocating engines. The saving in coal by its use was stated to amount to between ten and thirty per cent. The leading features of the Schmidt superheater appeared to be twofold. Firstly, and most important, was the placing of the superheater tubes in close proximity to a boiler surface, as in the case where a single convolution of superheater tube lay within the fire tube of the Scotch boiler, the great advantage being that the superheater tube was saved from burning and destruction by the rapidity of its radiation to the contiguous boiler surface. This very valuable feature seemed not to have been sufficiently recognized. The second important feature was that of high steam velocities, so as to scour out the tubes, and this was shared by many other superheaters, which were also giving excellent results. To judge from the steady progress of superheat on ships, it might safely be assumed that its use would extend further, and that a rate of fuel consumption per H.P. would be attained in large power installations more closely approximating to that of the internal-combustion engine. The abnormally high speed of the propellers of Diesel-engined vessels involved a very considerable loss in propulsive efficiency as compared with steam-driven machinery, whether reciprocating or geared turbines, and that that difference might be of the order of ten per cent. to twenty per cent., depending on the stern lines of the vessel.—"Marine Engineer and Naval Architect."

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## TURBINE MACHINERY FOR SHIP PROPULSION.

Whilst there has been a considerable development in the application of the steam turbine for ship propulsion, the question of the most suitable turbine arrangement on board ship appears to be no nearer finality than it was 10 years ago, when the first cross-Channel steamer was put into service by the South-Eastern & Chatham Railway Company. The then more or less standard of one high-pressure cylinder and two low-pressure cylinders that has been so very widely adopted in mercantile work seems now very likely to be displaced by the twin-screw gear turbine arrangement, just as the cumbrous system of installing cruising turbines has been suppressed in warships. In discussing this matter in one of its recent issues, our contemporary "The Engineer," remarked that in the adoption of gearing would seem to lie a future predominating influence on turbine design compared to which even the all-important method of placing turbines in series on separate shafts can hardly compare, although the latter was an invention which must have enormously hampered the development

of other types for marine purposes, in view of the exceptional advantages it offered for reducing the weight and improving the economy of the reaction turbine. But for larger vessels than any yet built, it appears likely that helical gearing will also be adopted, and if this takes place it is indeed hard to imagine what the limits to size may be. True, for the larger and less numerous vessels of great power the change may not take place at once, although it must be remembered that there are at the present time under construction many liners of great size having geared turbines apart from the high-powered Admiralty vessels in which gearing is now being adopted.—"Shipbuilding and Shipping Record."

## CONDENSING APPARATUS.

### RATES OF HEAT TRANSMISSION.

A considerable number of experiments have been carried out in recent years on the heat interchanges which take place within a surface condenser, and though the theory of condensation is by no means in a satisfactory state, some general results have been secured which make it possible to fix, with reasonable precision, the extent of surface and the supply of circulating water required in practice. Thus it is commonly assumed that with clean tubes the rate of heat transmission may be about 500 B.t.u. per square foot per hour per degree of difference in the temperature of the fluids between which the interchange of heat is being made. With dirty tubes the rate may fall to about one-half the above figure, and, as is well known, the presence of air in the steam greatly affects the rate of heat transmission. The actual rate of transmission per degree of difference in the temperature of the walls of the tube, however, is, of course, much greater, being, with brass tubes of the usual thickness, at the rate of about 12,000 B.t.u. per square foot per hour per degree of difference of wall temperatures.

In condenser work the pressure and temperature of the steam are both low; but in such articles as evaporators and the like, fairly high steam pressures may be used, and but few results of observations made under such conditions of operation have so far been made public. A special value attaches, therefore, to the results lately published of a series of interesting experiments on rates of heat transfer which have recently been carried out at the Agricultural Experimental Station of the Louisiana State University by Professor E. W. Kerr and Mr. A. J. Isacks. The primary object of the investigation was to determine the best conditions for the operation of the double and triple-effect vacuum pans used in the preparation of sugar. The general type of evaporator used in these tests is illustrated in Fig. 1, which represents one of the three elements which are operated in series to constitute a complete triple-effect evaporator. It consists, it will be seen, of what is essentially a surface condenser, with the tubes arranged vertically. The steam which effects the evaporation of the juice inside the tubes is supplied to the exterior, and the condensate drained off at the bottom. A "vapor space," 10 feet high, extends above the upper tube plate; but the level of the liquid to be evaporated is generally kept considerably below this upper tube plate. The tubes used were in all cases of copper 18 B.W.G. thick. They were in some of the experiments 1⅜ inches in diameter by 24 inches long, whilst in another set of experiments they were 4 feet long and 2 inches in diameter. The number used in this case was 39, whilst of the smaller tubes 80 were used in one set of experiments and 66 in another, some of the inner tubes of the set of 80 being in this case replaced by a downcomer, 6 inches in diameter, extending between the top and bottom tube plates. The only

Fig. 1.

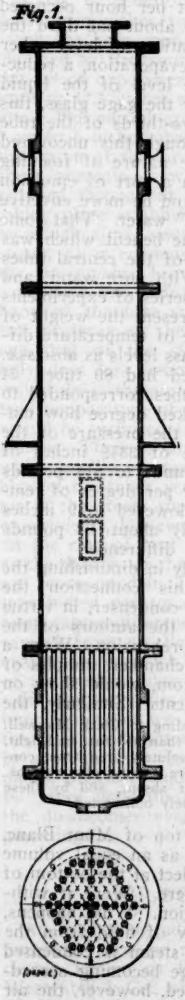


Fig. 2.

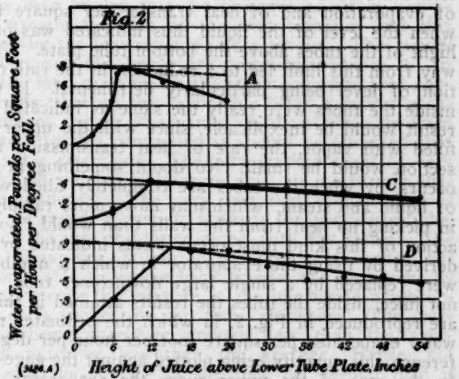

Water Evaporated, Pounds per Square Foot per Hour per Degree Fall.
A
C
D
0 6 12 18 24 30 36 42 48 54
Height of Juice above Lower Tube Plate, Inches

Fig. 3.

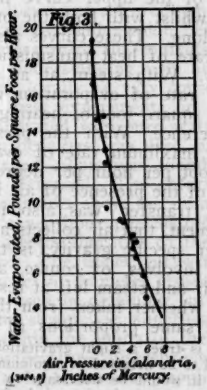
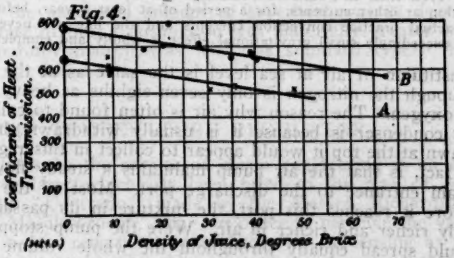



Fig. 4.



circulation within the tubes was provided by convection currents. A gage glass was fitted by which the hydrostatic level of the juice under evaporation could be determined, and it is very noteworthy that the highest rates of evaporation and of heat transfer per square foot per hour occurred when the level of the liquid thus indicated was only about one-third the hight of the tubes above the bottom tube plate. Variations of level either way from this limit led to a reduction in the rate of evaporation, a reduction of level being particularly detrimental. If the level of the liquid inside the tubes were really the same as indicated by the gage glass, this result would be inexplicable, since with the upper two-thirds of the tube filled with vapor, the rate of heat transmission through this uncovered section would be small. No doubt something in the nature of foaming occurs, by whch the tubes are completely filled with a sort of emulsion of liquid and steam, which may move more rapidly and be more effective in picking up heat from the walls than would "solid" water. That some action of this kind must occur seems indicated by the benefit which was derived by using the evaporator in which a number of the central tubes were replaced by a single large downcomer tube. With pure water, and not juice, inside the tubes the results attained in this series of experiments are reproduced in Fig. 2, in which the ordinates represent the weight of water evaporated per square foot per hour per degree of temperature difference, this quantity being plotted against the gage-glass levels as abscissæ. In the case of the upper curve the evaporator used had 80 tubes, 24 inches long, whilst with both the other curves the tubes corresponded to tubes 4 feet long. These curves also show in a marked degree how materially the rate of heat transmission is affected by the pressure of the steam supply. With steam at an absolute pressure of 33¾ inches of mercury, the rate of evaporation was about 7¾ pounds to 8½ pounds of water per square foot of heating surface per hour per degree of temperature difference. With the pressure of supply lowered to 9 inches absolute, the maximum rate of evaporation was only about 4.2 pounds per square foot per hour per degree of temperature difference.

The effect of the presence of air in the steam supply in diminishing the rate of heat transfer was also investigated. In this connection the curious statement, that air collects at the bottom of a condenser, in virtue of its greater specific gravity, is twice repeated by the authors of the paper. This involves, of course, a complete misapprehension. Were a mixture of air and steam left at rest within a closed chamber, samples of the mixture, whether taken from the top or the bottom, would show on analysis* the same proportions of the two constituents. Similarly, the

*There is, it is true, a slight gravitational effect, but, according to Clerk Maxwell, this is too small to be detected in a column of mixed gases less than 300 feet in height, and even then it would be necessary to keep the column absolutely free from convection or other currents, for a period of at least a year, before making observations. In actual practice convection currents and the like are never absent, and by these the exceedingly small gravitational effect is wholly and completely obliterated.

constituion of air at sea level is the same as at the top of Mont Blanc, although the nitrogen is only seven-eighths as heavy as an equal volume of oxygen. The reason why air is often found to collect at the bottom of the condenser is because it is usually withdrawn there. Were it withdrawn at the top it would appear to collect in this region. What happens, in fact, is that the air pump maintains a steady flow of fluid from the steam entrance to the discharge port. Most of the steam is condensed before it reaches this port, the mixture in its passage becoming accordingly richer and richer in air. Were the pump stopped, however, the air would spread equally throughout the whole volume of the chamber by diffusion, and after the lapse of a few hours the local concentration would have entirely disappeared. The richness in air of the mixture at the air-pump suction is thus not in any way dependent on the fact that the

density of air is greater than that of steam, but arises solely from the combined effect of the condensation of the steam and the establishment of a current from the steam inlet to the discharge port.

It has, of course, long been known that the presence of air in a condenser diminishes very greatly the rate of the heat transfer, and this result has been confirmed by the experiments under review. The results obtained by Professor Kerr and Mr. Isacks are represented in Fig. 3. In this set of experiments the total pressure on the steam side was kept constant at 17.5 inches absolute, that on the vapor side being 8 inches absolute. The partial pressure of the air was varied on the steam side from 0 up to 6 inches absolute. In this connection it should be observed that the action of the air is not a direct one. The very marked effect of very small percentages would be unintelligible were this air uniformly distributed throughout the condenser. The very fact that condensation is taking place establishes, however, a current towards the condenser tube, by which air is swept in towards the latter along with the steam. The steam, as it condenses, disappears, falling to the bottom as water, but the air remains, being only able to escape back against the in-current by convection and diffusion. Hence the mixture in the immediate neighborhood of the tube surface is richer in air than elsewhere. The partial presence of the air being therefore greater here than the average, that of the steam must be less than the average. Hence, not only does less steam reach the surface of the tube per second than it would do in the absence of air, but it does so at a lower temperature, the true temperature of the steam in the act of condensation being less than would be surmised from the average pressure of the contents of the condenser. There is thus no difficulty in understanding why an air content which on the average is very small may locally be very considerable, and blanket the tubes very effectively. In these experiments on the effect of air in the steam supply the liquid evaporated was pure water. In other experiments, however, juice of varying densities was used, and the effect on the rate of heat transmission observed. This was found to diminish as the density of the liquid evaporated increased. The observations made are represented in Fig. 4, where the ordinates show the number of B.t.u. transferred per square foot of surface per hour per degree of (Fahr.) temperature difference. The abscissæ are the densities of the juice which ranged from 8.8 Brix up to 67.3 Brix. The upper curve relates to an evaporator in which a downcomer 6 inches in diameter replaced a number of the central tubes of the evaporator used in obtaining curve A.

In another series of experiments the steam was superheated, and the important point was established that within the limits of the experiments the B.t.u. transmitted per degree of temperature difference was unaltered by superheating. In these experiments the steam was supplied at atmospheric pressure, and the maximum amount of superheat was 33 degrees F. Here again the rate of heat transmission was materially greater with the downcomer evaporator than it was with the standard type.—"Engineering."

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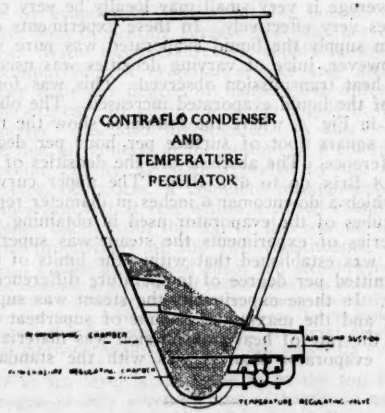
## AIR-PUMP TEMPERATURE REGULATOR.

An interesting addition to the science of steam engineering is an air-pump temperature regulator which has been fitted to the type of condenser known as the "Contraflo," made by the Messrs. Richardsons, Westgarth & Co., of Middlesbrough, Eng. In order to perceive the value of this development a few remarks on the theory of condensing as applied to marine practice may be advisable. Except in passenger boats of large power the standard practice on a steamship fitted with reciprocating engines is to withdraw air and water from the main condenser by a single air pump driven by levers from the main engines. In order to re-

move air from a condenser by an ordinary reciprocating air pump of the barrel and bucket type, it is fundamentally necessary for the temperature within the barrel to be less than the temperature within the condenser.

This will be seen when it is remembered that the temperature at which the water boils depends on the pressure to which it is exposed. At atmospheric pressure the boiling point is 212 degrees and at 28 inches vacuum it is 101 degrees, and so on. If water at 101 degrees F. were taken into an air pump it would be impossible to produce a greater vacuum than 28 inches, as any attempt to do so will make the water boil and the barrel would fill with steam. Therefore, in such a case, the vacuum in the condenser must be less than 28 inches or else no air would flow from the condenser to the air pump. When the condenser and the air pump are equally hot the withdrawal of air is impossible, however large the air pump may be, because the pressure in the condenser and in the pump are equal. If, however, the pump is made colder than the condenser the pressure in the pump falls and air flows into the pump. The greater the difference in temperature between the pump and the condenser the greater is the difference in pressure and the greater will be the quantity of air the pump can withdraw. Obviously, therefore, the withdrawal of air and water by one pump introduces a compromise, for in order to withdraw air and allow the condenser to do its work effectively the air pump has to be relatively cold, but in order to obtain high thermal efficiency the temperature of the feed water must be relatively hot. In order to conserve heat the water passing through the air pump must be as hot as possible relatively to the temperature in the condenser.




In order to adjust these conditions as far as ever possible the air-pump temperature regulator has been introduced, the actions of which will be seen from the illustration. The bottom of the condenser is provided with a divided-off receiver which is always kept filled with condenser water. Above this receiver is a shallow chamber in which the air is devaporized as it flows towards the air-pump suction pipe. The devaporizing chamber is divided from the condensing department by one or more diaphragms on to which the water of condensation falls and from which it passes either to the air pump or to the water-charged receiver or partly to both.

The flow of the water is regulated by a change valve and the inlet pipe of the receiver provides sufficient head to cause the water to circulate. If when the feed water passes direct from the condenser chamber to the air-pump suction pipe the air pump is able to prevent any accumulation of the air in the condenser, then the air-withdrawal capacity of the pump is obviously sufficient. If, however, the pump cannot prevent air accumulation in the condenser the vacuum falls. Under such circumstances the air-withdrawing capacity of the pump can be immediately increased by lowering its temperature, and this can be effected by passing some water from the condensing chamber through the water-charged receiver. In this way the air-withdrawing capacity of the pump can be increased at will and a vacuum maintained in a way which would be impossible by any other means, even if two ordinary-sized pumps were fitted instead of one. It is temperature, and not size, which controls the air-withdrawing capacity of an air pump. For this reason a higher vacuum can be maintained in a steady manner in all seas when the temperature regulator is used in combination with a single air pump withdrawing both air and water than has hitherto been possible even with an air pump of abnormal size.

The great importance of maintaining a good vacuum under all conditions is so thoroughly understood by marine engineers that further stress need not be laid on the important nature of this development, which is of first-class importance for any vessel where steaming economy has to be realized.—"Marine Review."

## BOILERS.

### SUPERHEATING MARINE BOILERS.

When Sir Charles Parsons, in addressing a meeting of the North-East Coast Institute of Engineers and Shipbuilders towards the end of last year, in a presidential capacity, touched upon the subject of superheating in marine boilers, he said that though superheating was not mentioned in the report of Lloyd's Register, yet it had quickly made great strides and was now employed in the mercantile marine to the extent of upwards of 1,000,000 H.P. on the Schmidt system alone, nearly the whole of this being in conjunction with reciprocating engines. The saving in coal was stated to amount to between 20 and 30 per cent. The leading features of the Schmidt superheater appeared to be twofold. Firstly, and most important, was the placing of the superheater tubes in close proximity to a boiler surface, as in the case where a single convolution of superheater tube lay within the fire tube of the Scotch boiler, the great advantage being that the superheater tube was saved from burning and destruction by the rapidity of its radiation to the contiguous boiler surface. This very valuable feature seemed not to have been sufficiently recognized. The second important feature was that of high steam velocities so as to scour out the tubes, and this was shared by many other superheaters, which are also giving excellent results. To judge from the steady progress of superheat on ships, it might safely be assumed that its use would extend further and that a rate of fuel consumption per horsepower would be attained in large power installations more closely approximating to that of the internal-combustion engine. The abnormally high speed of the propellers of Diesel-engined vessels involved a very considerable loss in propulsive efficiency as compared with steam-driven machinery, whether reciprocating or geared turbines, and that this difference might be of the order of from 10 to 20 per cent., depending on the stern lines of the vessel.—"Shipbuilding and Shipping Record."

## INFLUENCE OF SOOT ON BOILER ECONOMY.

By H. D. Fisher.

The writer has been engaged in boiler-plant work for a number of years, and has made a careful study of the accumulation of soot and methods for removing it, also its effects, both by boiler tests and inspections of boilers, when they were out of service, on which various methods of cleaning were used.

Recently the writer ran tests on a boiler equipped with a tube-blowing device to determine its effect on boiler economy. The water-tube boiler was 400 H.P. capacity, 21 feet wide by 9 feet high, 4-inch by 18-foot tubes, three 36-inch by 20-foot 2-inch drums and a superheater with 439 square feet of surface. It was hand fired with a mixture of Pennsylvania bituminous coal and screenings in the proportion of about one to three, on a rocking grate 12 feet wide by 7 feet 4 inches long, or 88 square feet area. Test *A* was run one week after the tubes had been thoroughly swept and the boiler operated as usual without blowing or cleaning the tubes in any way. Test *B* was run two weeks after the test *A* and one week after the tubes had been swept, the boiler being operated as usual and the tubes blown daily. The tests were of two-hour duration and were started and stopped by the alternate method.

| Number of Test. | A | B |
|---|---|---|
| Steam pressure, gage, pounds | 130 | 131 |
| Temperature feed water, degrees | 208 | 193 |
| Temperature gases leaving boiler, degrees F | 437 | 440 |
| Temperature superheated steam, degrees F | 424 | 435 |
| Draft at boiler damper, inches water | 0.50 | 0.46 |
| Weight coal as fired, pounds | 13,207 | 14,604 |
| Per cent. moisture in coal | 2.74 | 1.54 |
| Total ash and refuse, pounds | 2,835 | 3,538 |
| Coal per square foot grate per hour, pounds | 15.0 | 16.6 |
| Total water fed to boiler, pounds | 111,419 | 121,499 |
| Water evaporated per pound coal, pounds | 8.43 | 8.32 |
| Per cent. rated horsepower developed | 88 | 98 |
| Heating value of coal per pound | 13,057 | 13,410 |
| Efficiency of boiler grate, per cent | 67.9 | 66.7 |
| Loss due to latent heat of water, per cent | 2.3 | 2.2 |
| Loss in products of combustion, per cent | 6.9 | 7.2 |
| Loss in air excess, per cent | 6.3 | 7.2 |
| Loss due to unburned carbon monoxide, per cent. | 0.6 | 1.2 |
| Loss due to unburned carbon in ashpit | 6.7 | 5.3 |
| Radiation and unaccounted losses | 9.3 | 10.2 |
| | 100.0 | 100.0 |

Neither from these tests nor the continuous operating results of the plant could any improvement in operating results be noted.

A particularly bad feature of all forms of soot-blowing apparatus which require a number of pipes piercing the boiler side walls, especially if they are installed after the boiler setting is erected, is the cracking up of the brickwork and the amount of air leakage this allows. In some tests on 302-H.P. water-tube boilers with hand-fired flat grates, burning a mixture of two parts Pocahontas to one part screenings, it was found, from gas samples taken over the fire in the first pass, that there was 11.6 per cent. $CO_2$, corresponding to 77 per cent. excess air, while in the back connection simultaneous samples showed only 5.8 per cent. $CO_2$, corresponding to 280 per cent. excess air, or that more cold air was leaking in through the boiler walls than was coming up through the grate; they were operating a part of the boiler as an air-cooled condenser.—"Power."

## FUEL-GAS ANALYSIS AND ITS ADVANTAGES.

By Mr. Arthur E. Jones.

It is gradually coming to be realized that no steam-raising plant which can lay claim to any importance is really completely furnished unless automatic $CO_2$ recorders are installed, and their favorable influence on efficiency is being so brought home to up-to-date engineers that it will probably not be long before they are specified in contracts in the same was as steam-pressure gages.

Before we proceed to investigate the reason for this, and see what information these instruments can give us and in what way they may be of assistance, it may be well to consider the general question of steam production, and in so doing we will devote our remarks particularly to the fuel mostly employed in this country for mill driving, and, up to the present at least, for marine purposes also, namely, bituminous coal, although they will apply in a large measure to other fuels also.

When we wish to produce steam in a boiler we must do so by passing sufficient heat into the water contained therein, and coal usually forms our most convenient medium, containing as it does a considerable amount of heating power in a comparatively small volume, and yet being easy to procure and safe to handle. The heating power or calorific value of any coal can be readily ascertained by one versed in the necessary methods, with the aid of a calorimeter and other apparatus, and in England it is usually expressed in British thermal units per pound of fuel, the British thermal unit, or B.t.u., as it is commonly called, being that amount of heat which is required to raise one pound of water one degree Fahrenheit in temperature.

The heating value of any given coal being then something definite and fixed, it might readily be thought that two firemen burning equal amounts of the same coal in the same boiler would always produce equal amounts of steam. Most of us know, however, from our own experience that such is not the case, and official tests carefully carried out have shown results varying very considerably. Since there is a reason for everything it may be well worth our while to seek for the reason of this, as our researches may help to bring home to us more forcibly some points to which we have hitherto not attached sufficient importance. Let us then commence.

When coal is being used in a boiler furnace for the purpose of steam production, only a part of its value can be transmitted to the water, and the remainder is lost. If, therefore, one of the firemen shows a higher efficiency, it is evident that since there is a definite amount of potential heat in the coal, he must be increasing the quantity usefully employed by reducing the losses to an equivalent extent. What then are these losses? They can all be accounted for under one or other of the following heads: (*a*) Coal not burned, or only partially burned, falling into the ashpit; (*b*) heat radiated from the furnace, boiler surfaces, and brickwork if any; (*c*) heat in clinker and ashes taken from the fires when cleaned out; (*d*) heat passing away up the chimney or funnel.

Now in the two cases which we are considering there will not be serious differences in the first three items, namely, the losses in the ashpit and clinker, and by radiation, and the reduction in loss must be looked for under item (*d*), the better fireman having conducted his management of the fires in such a way as to allow less heat to pass away up the chimney or funnel.

We have now narrowed down the issue considerably, and can make further progress by investigating the nature of these chimney or funnel losses, the greater or less amount of which can account for such considerable differences in boiler efficiency. This will take some little time, and we can best begin by considering the composition of coal—the fuel

which we use for producing the necessary heat—and the process of combustion.

Coal is composed of oxygen, nitrogen, ash, sulphur, hydrogen and carbon, of which the two last named are the valuable combustible elements—carbon always being by far the greater in amount. In order that their potential heat value may be developed and converted into sensible heat—that is to say, in order that they may burn and so give up their heat—they must be brought into close contact with a sufficient volume of oxygen at a temperature high enough to bring about their combination. It is important to remember that, in order to produce complete combustion, all these three conditions must be present at one and the same time—sufficient oxygen, thorough mixing, and a temperature high enough.

When we say sufficient oxygen we are by no means speaking vaguely, or in the dark, for we know exactly what amount is required for a given weight of carbon or hydrogen, and fairly accurately what amount of heat is derived from their complete combustion, and, since we know what percentage oxygen forms of atmospheric air, we can equally well tell what amount of air is needed for these processes.

One atom of carbon combining with one atom of oxygen forms CO—carbon monoxide—and yields heat at the rate of 10,230 B.t.u. per pound of carbon. Out of this, however, 5,810 B.t.u. are absorbed in the work of changing the carbon into the gaseous form, leaving only for our use a net available heat of 4,420 B.t.u. The addition of a second atom of oxygen converts this CO into $CO_2$—carbon dioxide—and gives further heat again at the rate of 10,230 B.t.u.—all available this time—since the CO is already in gaseous form—thus making a total net heat of 14,650 B.t.u. per pound of carbon.

There is in all coals, as already stated, far less weight of hydrogen than carbon, but its heating power per pound is considerably higher, viz: 62,100 B.t.u. The available heat for steam raising in a boiler is, however, less than this for the following reason: The hydrogen combined with oxygen in the proportion of two units of the former to one of the latter, forming aqueous vapor, $H_2O$, in weight nine times as much as the original hydrogen. Since the flue gases pass up the chimney at a higher temperature than 212 degrees F., boiling point, this $H_2O$ will escape as steam, carrying away its latent heat, which is 967 B.t.u. per pound. Nine pounds will therefore, rob us of 8,700 B.t.u., leaving a net available heat of 53,400 B.t.u. per pound of hydrogen. (These values of carbon and hydrogen are according to Bryan Donkin—some other authorities give values slightly different.)

From these investigations we clearly see the importance of introducing to the boiler furnaces enough air to provide sufficient oxygen to fully develop the potential heat in the combustible elements of the coal, since otherwise we may sustain considerable losses.

To take the carbon first, any particles for which no oxygen is provided will carry away their whole heating power to waste and leave the chimney top in the visible form of smoke, thus, in addition, incidentally creating a nuisance. Those portions which secure to themselves one unit of oxygen per unit of carbon will give up, as we have seen, less than one-third as much heat as they would under perfect conditions, and only from that part of the carbon which unites with sufficient oxygen to form $CO_2$ do we derive the full benefit of its heating power.

Most authorities agree that, when the air enters the ashpit of a boiler and passes through a mass of incandescent fuel in the furnace, CO is first formed—that is, the product of incomplete combustion—and, if this gas afterwards meets with a further stream of air before it has been cooled down below the temperature necessary for combustion, it is converted into $CO_2$ the product of complete combustion.

With regard to the hydrogen, we need only say that, whereas its com-

plete combustion produces a comparatively large amount of heat, this heat is entirely lost if a deficiency of air prevents its meeting with the requisite oxygen, the hydrogen then passing away absolutely unconsumed.

In order, therefore, to prevent any likelihood of these various losses through incomplete combustion, we shall be wise to take care to admit plenty of air. The next point to consider is what the consequence will be if we go to the other extreme and admit a great deal too much air, and we shall soon find that we then sustain even greater losses in two different ways.

In the first place, a smaller proportion of the heat in the gases of combustion will be usefully employed in heating the boiler water, and more will be wasted up the chimney, on account of the cooling effect of the superfluous air, and the cause of this is very plain. As a ball will run downhill more or less rapidly according to the steepness of the hill, so heat will be transmitted through a boiler plate more or less quickly according to what is called the heat gradient, that is, the difference in temperature between the hot gases on the one side and the water on the other. Therefore the furnace gases, reduced in temperature, will, during the short time whilst they are being rushed along in contact with the boiler-heating surfaces, succeed in passing a smaller proportion of their heat into the water than would have been the case had so much excess air not been admitted, and, therefore, at the finish, more heat will, of course, be left to go to waste up the chimney.

In the second place we must remember that the air which we admit to the furnace at atmospheric temperature contains in addition to the useful oxygen the inert gas nitrogen, to the extent of more than three-quarters of its entire weight. Now the whole of this nitrogen, and also the gases of combustion, leave the chimney at a greatly-increased temperature, carrying away a large amount of heat, and it is very evident that the larger the excess of air we admit, the greater will be the volume of the escaping gases, and the greater, therefore, the amount of heat carried away to waste at any given temperature.

Now, then, we are better able to understand how it is possible for the one fireman, with the same amount of coal, to produce more steam than the other one, for he evidently manages his air supply better, as compared with the coal burned, either through knowledge or instinct or good luck. The less fortunate fireman sends more waste up the chimney, either through admitting too little air, thus losing a part of the value of his coal through incomplete combustion, or else through admitting too much air, whereupon the increased volume of gases carried away a larger percentage of heat.

Staying now for a moment to review the position we appear to be in somewhat of a difficulty, and to find ourselves between Scylla and Charybdis, for if we have a shortage of air we suffer loss, and if, on the other hand, we have an excess of air we may suffer even greater loss. Evidently, the only way to secure economy is to avoid both extremes, and to so arrange matters that the air supply is just about sufficient for the coal burned. But how are we to know when this is the case, for neither the air nor the gases can be seen, unless the latter be smoky, and there are so many points to be considered? The kind of coal, its size, the thickness of the fire, the spaces between the firebars, the width of the firebars, the openings in the furnace door, the strength of the draft—all these things will affect the fires have combined in one common stream. A small pipe is run from the uptake of each boiler which the recorder is to control to a spot close to the instrument, which must, of course, be located in a clean place, the final connection being made by means of a short length of rubber pipe which is temporarily joined by some simple and convenient method to whichever line of piping it is desired to test.

In this way a chief engineer—whose responsibilities are widespread, and

who has too many matters to think about to devote overmuch consideration to any particular one—can, by the expenditure of only a reasonable amount of time and attention, have clear and lasting evidence of the efficiency with which his boiler furnaces are being managed. First of all he will see whether the general methods which are followed are fairly satisfactory—then by switching over at intervals from one connection to another he will be enabled to tune up the various boilers in rotation, and test any individual one at will.

The feeling that they may be under test at any time tends to keep the stokers' work at a higher level, and the fact that the visible results can be inspected and discussed helps to increase the intelligent interest of all concerned. A careful study of the records, side by side with the knowledge of the conditions existing at the time, may often yield much information and result in the discovery of faults which might otherwise have remained unknown. As has already been mentioned, a number of factors exercise an influence on the furnace efficiency, but a faulty state of things with regard to any single one of them will be quite evident upon examination of the $CO_2$ record, which will point to the need for an alteration in one direction or the other. The explanation of this is that every one of these factors affects in some way or other the flow of air for purposes of combustion, and, as we have seen, the whole question of efficient combustion rests on the correct proportioning of air supplied to fuel burnt. The $CO_2$ record, therefore, gets straight to the point by showing the result of all the factors combined, and so may well be called the pulse which shows the healthy condition or otherwise of the whole system.

One aspect of the matter which we have so far passed over is the bearing which $CO_2$ recorders have upon smoke prevention. Many steam users are quite convinced from practical experience that a smoky chimney is more economical than a clear one, and unfortunately the conditions usually existing appear to justify their belief, although it is only a case of a half truth being worse than a lie.

We have shown that the highest efficiency will exist when a slight excess of air is admitted to the furnace, thus bringing about complete combustion, when the flue gases will be clear, but that, if we lessen the air supply, the combustion will be incomplete and smoke will ensue with resulting losses. If, on the other hand, we increase the air supply more and more the gases will remain clear, but the loss of heat up the chimney or funnel will gradually become greater and greater, until at last it may far exceed that caused by incomplete combustion accompanied by smoke.

The half truth then, which is more dangerous than a lie, is the fact that a chimney, smoky through shortage of air, may be less wasteful than one clear through large excess of air, although one clear with a slight excess gives the very highest efficiency.

Now it is scarcely possible to distinguish between these two conditions of clearness without the help of a $CO_2$ recorder, the chart from which will enable one to steer between the two extremes and keep near to the smoke line, without, however, passing over it to any serious extent. A safe rule to adopt with marine boilers is to endeavor, when sampling gases from the spot suggested, namely, the uptake, to produce a record varying from, say, 10 to 13 per cent. of $CO_2$. It is true that we stated that 15 per cent. represented practical perfection, but it must be remembered that there will be some slight unavoidable inleak of air round the smoke-box doors which will dilute the gases in proportion to its volume; care must, of course, be taken to keep it as low as possible, but it is extremely likely that it will be sufficient in quantity to bring down the advisable maximum, at least, from 15 per cent. to 13 per cent. At 13 per cent., then, under such conditions, one may very well anticipate that smoke will not be produced to any extent, and between that point and 10 per cent. the loss through excess air will not be unduly great.

The records will help to bring about a healthy rivalry between the various shifts, and it might me of advantage to devise a system of bonuses to encourage the stokers to show good results. Furthermore, the question appeals to them personally when they once realize that the more efficiently they burn the coal the less they have to throw on.

A consideration of all these points is of great importance in connection with boiler practice anywhere, but for various reasons it appears to be so to a superlative degree where installations on board ship are concerned.

In the first place the price of coal, which has to be shipped in different parts of the world, is frequently very high, and with the price the value of economy, of course, increases in proportion. Now the amount of coal burned throughout the course of a year on any vessel is comparatively very large, so that quite a small percentage of saving through increased efficiency would total up to a considerable item annually.

But there is another point which on some boats would be even more advantageous than that, and which a chief engineer was good enough to mention to the writer.

Take the case of a fast liner on which the rate for cargo was very high. If the proper use of $CO_2$ recorders permanently raised the efficiency of the boiler furnaces, a less quantity of coal would suffice for each voyage, so that not only would there be the resulting economy in coal, but also valuable space which had formerly been reserved for fuel could be utilized for cargo, and the return on the capital expenditure on recorders would be exceedingly large.

With regard to the beneficial influence of the recorder on the smoke question, it is obvious that, apart from economy, the avoidance of any considerable amount of smoke from the funnels would be desirable, especially on passenger boats.

Then again the recorder has a very direct bearing on the serious question of leaky tubes in marine boilers. The more irregular the firing the more extreme and rapid will be the variations in temperature, resulting in undesirable stresses through expansion and contraction. But irregularity in firing would be accompanied by irregularity in the $CO_2$ record, which would plainly show what was happening, and call attention to the need for alteration in the methods of the stokers.

Seeing that the proportion of the heat value of the coal carried away by the escaping flue gases naturally varies with both their temperature and their volume, and the latter, as we have seen, bears a direct proportion to the percentage of $CO_2$ contained therein, a knowledge of these two factors—the temperature and the $CO_2$—will provide a basis for a calculation of the losses at any given time. A very useful chart worked out on these lines for ready reference is given in Bryan Donkin's "Heat Efficiency of Steam Boilers," and a glance at it will show in a very graphic manner what a serious thing it is, from the point of view of economy, to work with an unduly large excess of air, for it will be seen how the losses increase at an ever-accelerating rate as the percentage of $CO_2$ sinks lower.

In connection with this chart, it is interesting to mention a series of tests made under practical working conditions for the purpose of comparison. The boiler in use was provided with a special furnace which placed the air supply most thoroughly under the control of the fireman, who was instructed so to manage his firing as to produce good and bad records throughout alternate days. The coal was carefully weighed for each test, and the water measured—the temperatures of the flue gases were taken and the average percentages of $CO_2$ reckoned. In each case it was found that the actual difference in coal used to evaporate a given quantity of water under the good and bad conditions corresponded very closely with the figures shown on the chart.

Take, for instance, the two days' charts of which $CO_2$ records are shown in the reprint from "Engineering" on the subject. On the first day the average $CO_2$ was rather less than 8 per cent., and on turning to the chart, we find that, seeing that the temperature of the gases was approximately 500 degrees F. above atmospheric temperature, the amount of the heat value of the coal being carried away was about 24 per cent. of the total. On the second day, when the $CO_2$ was just over 15 per cent., this loss is found on the chart to be only 13 per cent., a reduction of 11 per cent. thus being shown.

The discrepancy between this economy of 11 per cent. on theoretical reckoning and the saving in coal actually observed was very small, for, whereas on the first day a certain quantity of water was evaporated by 100 pounds of coal, on the second day 89½ pounds sufficed for the same duty. In passing, we may notice that the run from 9 o'clock to 1 o'clock on the second day shows most excellent, in fact, practically perfect results —evidence of a rapid cleaning of the fire being noticeable where one short stroke occurs. Earlier in the day, however, and in the afternoon, the anxiety of the stoker to produce a good record led him to extremes, and he cut down the air supply too much, for 17 per cent. of $CO_2$, even at the downtake of a Lancashire or Cornish boiler, is unduly high, for any advantage which might be looked for from the last 1 or 2 per cent. would in most cases be more than counterbalanced by the loss resulting from incomplete combustion through shortage of air. It is just possible that in this particular instance no such loss through incomplete combustion did really occur, for, as already mentioned, a special boiler furnace was in use, and this ensured a more than usually thorough mixing of the air with the gases emitted from the fuel, so that it was fairly safe to work with only a small excess of air above that theoretically required. It is, however, almost invariably wise to keep down to such figures as those given earlier in this paper.

From all that has been said it would appear to be quite justifiable to claim for the automatic $CO_2$ recorder a foremost place among those instruments which are recognized as useful aids to the steam engineer. It cannot now be regarded as being merely of theoretical interest, seeing that one can point to a combine in this country which has 35 distributed throughout its various works; power stations abroad with 14 and 18 respectively, and a well-known Steamship Co. which has installed these instruments on all the seven liners of its fleet after making a lengthy trial on one.

Like all other recording instruments the $CO_2$ recorder cannot of itself accomplish anything, but at least it will tell during the whole of the time when it is in operation a plain, unvarnished tale of what is happening in the department which it has to control, and enable the engineer to see whether all is well, or whether conditions are such as to need attention and improvement. Upon due consideration he will surely feel called upon to investigate its merits and possibilities thoroughly and practically, for without its clear and constant guidance how can he hope to reach and maintain the highest level of efficiency?—"The Steamship."

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## COMBUSTION OF OIL FUEL IN MARINE BOILERS.

At the last meeting of the Institute of Marine Engineers a paper entitled "Working Arrangements for the Combustion of Oil Fuel in Marine Boilers" was read by Mr. James S. Gander. The paper dealt with two systems of oil burning, viz: the steam with oil and the direct-pres-

sure systems, and the author's description of the latter system shows that the direct-pressure system is so called because the fuel is injected into the furnace by pressure from a fuel pump through a burner which is pierced at the end by a very minute hole. The fuel is heated and filtered before it arrives at the burner.

With the direct-pressure system it is very necessary to drain off water and filter the oil before it passes to the burners. If proper attention be given to this matter no trouble will be given by burners choking or by defective combustion in the presence of water. It is remarkable, he says, how long these burners will run without attention, considering that the orifice is so small that little more than a pint passes through in 1 minute under a pressure of 30 pounds per square inch. A special needle is supplied for cleaning the orifice in the burners, and a testing apparatus enables the engineer to test the burners before fitting them in position on the furnace. The simple attachment of burners to the distribution boxes renders the matter of cleaning or renewing a very easy job. The triple thread which is cut at the end of the internal needle imparts a spiral motion to the passing oil which tends to keep the orifice scoured and also assists in the atomization.

The temperature at which the oil will give the most economical results depends upon its nature and gravity. An average mixture of fuel oil will require to be heated to about 160 degrees F., Tarakan crude to about 120 degrees and some of the heavier European residue to as much as 240 degrees. All other things being equal, the correct adjustment of temperature and pressure are most important items in the efficient working of the direct-pressure system. This is a very simple matter. If the oil be too cold the spray will enter the furnace without igniting; if too hot the same will ignite as it leaves the orifice. With the correct temperature the flame should appear at little more than 1 inch from the burner. The best results are obtained with a fuel pressure of from 30 to 40 pounds. The color of the flame gives the same indications as in the steam and oil system. With double-ended stokeholds and consequently long leads of fuel pipes, a small additional heater may be fitted to the fuel line most remote in order to maintain an equal temperature to all the supply. If the piping is arranged so that an efficient circulation of the fuel takes place this is not necessary.

When raising the temperature of the fuel oil on direct-pressure systems the exhaust valve of the heating coils should not be opened too much if it is required to heat the oil to a high degree. If the valve be almost shut down the pressure in the coils will be nearly the same as the initial supply with a correspondingly high temperature, and the heating effect can then be regulated by the amount of opening of the steam valve. On the other hand, if the exhaust be too much open the pressure in the coils will be nearly the same as that of the exhaust line, or almost atmospheric, with a temperature which will not serve to heat the oil beyond 200 degrees F.

In extremely cold climates and with heavy-grade fuel a little difficulty may arise owing to the thickening of the fuel interfering with the working of the pump. This is provided against by the addition of a small heating coil to each of the suctions in the bunkers, so that when occasion arises a little steam may be passed through, which has the effect of raising the temperature of the surrounding oil.

Comparing the steam with oil and the direct-pressure system, the advantages are greatly in favor of the latter, especially for marine purposes. Although the steam system is very simple and easily worked by firemen of average intelligence, and the cost of installation must be comparatively small, yet the direct-pressure system is far more economical of fuel. Of direct-pressure systems, the Meyer Smith, by reason of the recent im-

provements which have been mentioned before, appears to hold the foremost place for economy and convenience.

Speaking from the practical standpoint of the marine engineer he considers that the following advantages are obtained from the use of oil fuel:

(*a*) The quantity of bunkers received can be accurately ascertained very quickly. Tallying while taking bunkers no longer necessary. No trimming.

(*b*) No cleaning of fires required—therefore no currents of cold air enter the furnace which may cause damage.

(*c*) Full and regular steam can be maintained without any trouble in all climates.

(*d*) Steam is under perfect control. On receiving an order by telegraph the engineer can increase or decrease the steam supply to the fuel pump without leaving the engine room.

(*e*) More perfect combustion with oil than with coal—hence greater calorific efficiency.

(*f*) The engineer, if necessary, is independent of his firemen.

(*g*) The boiler tops, bilges and all engine-room and stokehold spaces can be kept in an absolutely clean condition, and there is no smoke or fumes to discolor paintwork or ashes to choke bilges.

(*h*) Combustion not affected by heavy weather. At times when coal firing might become impossible, the oil-burning vessel can carry on as under normal conditions, and the bilges are always clear in the event of a washout.—"Shipbuilding and Shipping Record."

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## THE FUTURE OF OIL FUEL.*

The forecasting of the future of oil fuel in general is a complicated problem, the difficulty of which is commensurate with its interest and importance. To a large extent the immediate future depends upon the evolution of the internal-combustion engine, and, as regards, the mercantile marine, upon the measure of success attending the efforts to construct satisfactory engines of the Diesel type of large cylinder capacity.

In this connection it must be borne in mind that the advent of the turbine engine, and the economies resulting from the use of superheated steam, together with other recent improvements in the efficiency of the steam engine, have rendered the task of the advocates of the internal-combustion engine more difficult. Moreover, the replacement of the turbine steam engine by the oil engine involves a return to the reciprocating type of machinery, which is in itself a ground of objection as being in a sense a retrograde step, for at present there seems no prospect of the evolution of a successful turbine oil engine.

The progress of the petroleum industry during the years 1901–1912 is shown in the following table, and in illustration of the sensitiveness of the industry and of the precarious nature of the relation of supply to demand it should be stated that the falling-off in 1905 was entirely due to the interference with Russian oil-field operations caused by labor disputes, for during that year there was a considerable increase in the output of

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*Abstract of an address by the President, Sir Boverton Redwood, Bart., D. Sc., delivered to the Junior Institution of Engineers on Dec. 5.

the American fields. The further decline in 1906 was contributed to by a smaller production in the United States.

| Year. | | Metric tons. |
|---|---|---|
| 1901 | (Redwood and Eastlake) | 22,160,701 |
| 1902 | " " | 24,879,030 |
| 1903 | " " | 26,285,270 |
| 1904 | " " | 29,500,000 |
| 1905 | (United States Report) | 28,945,795 |
| 1906 | " " | 28,643,410 |
| 1907 | (Redwood and Eastlake) | 35,729,238 |
| 1908 | " " | 38,220,438 |
| 1909 | " " | 40,070,832 |
| 1910 | " " | 43,986,531 |
| 1911 | " " | 46,431,404 |
| 1912 | (David T. Day) | 47,276,725 |

The chief sources from which the world's supplies of petroleum were obtained in the years 1911 and 1912 and the quantities which they respectively contributed are specified in the following tables:

*World's Production of Petroleum in* 1911.

(Redwood and Eastlake).

| Country. | Quantity. Metric tons. | | Percentage of total. |
|---|---|---|---|
| United States— | | | |
| Appalachian field | 3,166,644 | | |
| Lima-Indiana field | 830,821 | | |
| Illinois | 4,175,605 | | |
| Mid-Continent field | 8,789,038 | | |
| Gulf | 1,557,008 | | |
| California | 10,817,918 | | |
| Other fields | 56,215 | | |
| | | 29,393,249 | 63.305 |
| Russia | | 9,032,532 | 19.453 |
| Mexico | | 1,873,547 | 4.035 |
| Eastern Archipelago | | 1,624,301 | 3.498 |
| Rumania | | 1,544,072 | 3.325 |
| Galacia | | 1,462,940 | 3.151 |
| British Isles | | 850,065 | 1.831 |
| Japan | 220,673 | | |
| Formosa | 218 | | |
| | | 220,891 | 0.476 |
| Peru | | 186,405 | 0.401 |
| Germany | | 142,992 | 0.308 |
| Canada | | 39,743 | 0.086 |
| Italy | | 10,390 | 0.022 |
| Hungary | | 3,000* | 0.007 |
| Other countries, including Egypt and Trinidad | | 47,277* | 0.102 |
| Total | | 46,431,404 | 100.000 |

*World's Production of Petroleum in* 1912.

(David T. Day).

| Country. | Quantity. Metric tons. | Percentage of total. |
|---|---|---|
| United States | 29,615,096 | 63.25 |
| Russia | 9,317,700 | 19.37 |
| Mexico | 2,207,762 | 4.71 |
| Dutch East Indies | 1,478,132 | 3.09 |
| Rumania | 1,806,942 | 3.70 |
| Galicia | 1,187,007 | 2.43 |
| India | 989,801 | 2.03 |
| Japan | 222,854 | 0.48 |
| Peru | 233,486 | 0.50 |
| Germany | 140,000* | 0.28 |
| Canada | 32,612 | 0.07 |
| Italy | 12,000* | 0.02 |
| Other countries | 33,333 | 0.07 |
| Total | 447,276,725 | 100.00 |

In estimating the relative importance of the various sources, it is necessary to take into account the ever-varying factor of the quality of the crude oil as governing the proportion of the various products obtainable. Not only does the oil of one country differ from that of another country, but in the same country we find many different descriptions of oil, sometimes in the same district, and even in the same spot the oil obtained at depth may present no similarity to that found nearer the surface. Cases have occurred in which the deeper strata, where the oil is more effectively protected against loss by fractional evaporation, have yielded a product containing many times the percentage of motor spirit present in the oil from the upper strata. The matter is further complicated by the continuous shifting of the chief centers of production from one country to another, or from one field to another in the same country. Thus, at the present time oil is being imported from Mexico into Russia.

### THE DIESEL MARINE ENGINE.

The principal feature of the oil-carrying trade of the Caspian Sea during the past few years has been the growth in the substitution of the Diesel engine for the steam engine as the motive power of the vessels employed. This has arisen from the far greater economy in fuel consumption of the Diesel engine system, for whereas the consumption of oil fuel for steam raising amounted to 3 per cent. to 3.4 per cent. of the oil carried from Baku to Astrakan in 1912, the Diesel engine consumption is stated to have been only ½ per cent.

The decline in the Russian oil output has been mainly due to the falling off in the production of the small area in the Baku region, on which the drilling operations have been largely concentrated.

Apart from the possibility of such modifications of the present method of carbonizing coal as would render an additional quantity of liquid products available for use as fuel, to which further reference will be made, an increased quantity of oil fuel could be obtained from Scottish shale, and very large supplies from the far more extensive deposits of similar shale which occur in France, Servia, Spain, the United States, South America, New South Wales, New Zealand, New Brunswick, Nova Scotia,

*Estimated.

Newfoundland, and elsewhere. It may also be found possibl the bituminous clay or Kimmeridge shale of Dorsetshire as a oil fuel.

### CONSUMPTION OF MOTOR SPIRIT.

In Great Britain alone the consumption of motor spirit, which in 1905 was only about 18,000,000 gallons, has increased to not far short of 100,000,000 gallons, the imports last year amounting to about 80,000,000 gallons. Of these imports, 46,000,000 gallons came from the Dutch East Indies, and 16,000,000 gallons from the United States. It is true that if we take the average proportion of motor spirit obtainable from the crude oil at 10 per cent., the amount imported last year represents only about 2½ per cent. of the world's production of this product, but it is a significant fact that whereas a few years ago the United States supplied the world, we only obtained from that country last year one-fifth of the quantity we required, and, with the rapid growth in the American home consumption, the export trade may soon reach the vanishing point.

Already, in fact, considerable quantities of motor spirit have been imported from the Dutch East Indies into the United States, and the utilization of the bituminous shales of the latter country as an additional source of mineral oils is being discussed.

The present situation is mainly due to the circumstance that the use of motor spirit in road vehicles has increased with startling rapidity—at a rate, indeed, which is presumably far greater than was foreseen by few, if any, of those responsible for the supplies of the fuel. In the United States alone the number of cars in use increased from 15,000 in 1902 to 590,000 in 1912, whilst the output of petrol in that country increased from 6,600,381 barrels to about 13,000,000 in the same period. Therefore, the number of vehicles using petrol had increased forty-fold, whilst the production of the fuel had only doubled.

### THE "CRACKING" PROCESS.

Attention is being directed to the conversion of the heavy oil into motor spirit by the operation of "cracking," or dissociation. This process was placed upon a scientific basis by the classical researches of Thorpe and Young, who showed in 1871 that by the action of heat under pressure solid paraffin is converted into hydrocarbons (paraffins and olefines), which are liquid at ordinary temperatures.

In one of the latest applications of the principle of "cracking" a suitable heavy oil and water are brought into contact with a large surface of iron or other metal in highly-heated retorts, and in these circumstances a considerable proportion of the oil becomes converted into excellent motor spirit.

A difficulty in connection with the increase in the output of motor spirit is that the average crude oil only yields a comparatively small percentage of a product of the desired volatility, and that an outlet must be obtained for all other products. For many years past the consumption of kerosene as an illuminating agent has been diminishing rather than increasing, and there has been keen competition among sellers, with concurrent reduction in the price obtained, and although the larger domestic use of petroleum stoves for heating and cooking has to some extent arrested the diminution of the burning-oil demand, the difficulty has been acutely felt. Public-service motor vehicles are now being driven in this country with a mixture of petrol and kerosene, and the latter product can be used alone with a suitable carburettor if the engine is started with petrol, which can be supplied from a small auxiliary tank, but the private consumer is justifiably reluctant to use kerosene as long as he can obtain motor spirit, even at an enhanced cost, for it is more difficult to get satis-

factory combustion with the heavier fuel, and in crowded streets there would probably be serious complaints of the character of the exhaust. Apart from this, it seems almost impossible to prevent the kerosene from gradually rendering not only the chassis, but even the interior of the vehicle unpleasantly oily and odoriferous.

The oil fuel which is chiefly used in steam raising is a product of a very different character. Crude petroleum of a low flash-point is largely used as a fuel in steam raising in the oil fields, but the oil fuel of commerce is usually required to be of comparatively high flash-point, and is either a distillate or more commonly the residue of crude petroleum from which the more volatile constituents have been removed by fractional distillation.

In the British mercantile marine the minimum flash-point is 150 degrees F. (Abel test), whilst in the United States it is 140 degrees F. by the Abel-Pensky or Pensky-Martens test, and in Russia, Roumania, and Germany it is 80 degrees C. (176 degrees F.).

### BURNERS FOR OIL FUEL.

For the use of such oil in steam raising, various systems have been from time to time patented. One of the simplest devices is the burner of Nobel, which consists of a number of shallow troughs, arranged in superimposed series at the front of the furnace, so that the burning oil flows by successive stages from the highest to the lowest, the flame of the burning fuel being swept into the furnace by induced draft passing between the troughs. With this apparatus it was found that 1 pound of Russian ostatki (petroleum residuum) evaporated 14½ pounds of water, whereas with the same boiler 1 pound of coal evaporated only 7 pounds of water.

The Wallsend Slipway & Engineering Company's burner has a nozzle with a conical orifice, from which the heated oil, under a pressure of 60 to 80 pounds per square inch issues as a spray in the form of a conoidal column of large diameter, with a rotary movement imparted by a helix on the valve stem, the resulting centrifugal effect materially assisting in the conoidal diffusion.

The improved construction of the furnace front is the principal feature of the Wallsend-Howden system of oil burning as applied to marine boilers. Each of the oil-splaying nozzles projects through a baffle plate, or the front plate of the casing, into an air trunk having lateral openings at its outer end. This air trunk projects concentrically with a second air trunk carried by the furnace front, and the annular space between the inner and outer air trunks is fitted with deflectors constructed in such manner as to give the air passing through the annular space a spiral motion. One advantage of this arrangement is that the openings for the supply of air are so disposed that direct radiation of heat from the furnace is prevented. When used with forced draft on Howden's system, with closed furnace front, an efficiency as high as 16.22 pounds of water evaporated from and at 212 degrees F. is claimed to have been obtained per pound of oil of a calorific value of 18,770 British thermal units, an efficiency of 83.9 per cent.

### MERITS OF OIL FUEL.

A lengthy experimental investigation of the merits of oil fuel was carried out by the United States Naval Liquid Fuel Board in 1902–3. Many forms of burner were tested, and the results obtained led the Board to the conclusion that the actual evaporative efficiency of a pound of oil as compared with a pound of coal might be considered to be in the ratio of 17 to 10, but that, taking various circumstances into consideration, the relative efficiency of oil and good steam coal from the naval standpoint of fuel supply in warships, might be regarded as in the ratio of 18 to 10.

In the use of oil fuel the construction of the furnace is of the highest

importance, for it is needful that the combustion should reach an advanced stage before the flame comes into contact with cooling surfaces. It will readily be understood that, with the limited furnace space available in steamships, and especially in warships, it was exceptionally difficult to devise conditions under which smokeless combustion could be obtained. The difficulty has been overcome by the adoption of a form of burner which gives a spreading flame commencing close to the jet, and by heating the oil, but chiefly by the provision of an effective air-heating furnace front.

The substitution of oil fuel for coal in steam raising is, however, a wasteful method of using this source of power, for whereas we do not ordinarily obtain with the steam engine more than 12 per cent. of the energy of the fuel in the form of work, in the case of the Diesel engine the return is as much as 37 per cent. Unfortunately, the difficulties attaching to the construction of engines of this type with large cylinders are great, but so much attention is now being given to the subject by engineers of the highest eminence that it may be confidently anticipated that the difficulties will be overcome. Meanwhile, the use of the Diesel engine of moderately high power on land is making steady progress, and much has already been accomplished in its application to marine work. The experience gained with such vessels as the *Selandia* and *Jutlandia* led to arrangements being made for the installation of Diesel engines in many other cargo boats, and it has been demonstrated that the fuel consumption is between 0.4 pound and 0.46 pounds per B.H.P.

The high thermal efficiency of the Diesel engine is due to the adoption of the principle of injecting the charge of oil into the air after compressing the latter, instead of mixing the oil and air and compressing the mixture, for the efficiency is proportionate to the extent of compression, and in internal-combustion engines of the ordinary type the risk of premature spontaneous ignition prevents the use of high compression. In the Diesel engine a pressure of as much as 32 atmospheres can be employed, and almost any description of mineral oil introduced in proper quantity into it at the moment of the completion of the pressure stroke is at once completely and most effectively consumed.

### OIL AND MARINE PROPULSION.

It is in the provision of power in ships that the superiority of the oil engine over the steam engine is most marked for, apart from the economy, there are great advantages in the saving of space, the distance which can be covered without replenishing the fuel supply, the reduction in labor, and the facility with which the oil can be taken on board; moreover, the engine can be started without previous preparation, and the consumption of fuel equally quickly arrested.

Some of these advantages attach to the use of oil as a substitute for coal, or even as an adjunct to coal, in warships, and to these may be added the valuable capacity for rapidly increasing the production of steam. In commenting upon the advantages of oil fuel, as demonstrated in the naval maneuvers carried out under Admiral May in 1906, Admiral the Hon. Sir E. K. Freemantle said: "When the stokers were pretty well worn out and had used all the coal that was nearest at hand, and much labor would be necessary to fetch coal and bring it to the fires, then they were able to use the oil fuel, and I believe I am correct in saying that the four ships which had oil fuel steamed away even from ships with nominally greater speed."

On the initiation of the late Lord Furness the cargo boats, the *Eavestone* and the *Saltburn*, were built and equipped respectively with oil engines and steam engines in order to furnish a practical comparison of the two

systems of propulsion. The former vessel was fitted with a set of Carel-Westgarth two-stroke cycle Diesel engines, and the latter with triple-expansion engines. Both vessels are 276 feet in length between perpendiculars and 40 feet 6 inches beam, and are otherwise similar. The capacity for fuel oil in the double bottom of the *Eavestone* is 156 tons, and the capacity of the coal bunkers of the *Saltburn* is 380 tons. The Diesel engines of the former vessel have four single-acting cylinders 20 inches in diameter by 36-inch stroke working upon the 2-stroke cycle at 96 r.p.m. The high-pressure and low-pressure air compressors are worked direct from the main engine; the high-pressure compressor for injecting the fuel oil being one of Reavell's patent three-stage compressors fitted at the forward end of the crank-shaft; the low-pressure is provided by two ordinary compressors worked by levers at the back of the main engines; the pumps for circulating water for the jackets, the bilges, and the oil-supply pump, &c., are also worked in the same way. There is an auxiliary Reavell air compressor for starting up and as a standby, which, together with the ballast pump and other auxiliaries, is steam-driven, and the necessary complement of air receivers, oil filters, &c., is provided. The winches, steering gear, &c., are the same as those of the *Saltburn*, and as steam is required at sea for the steering gear and whistle, and for heating the cabins, two auxiliary boilers, each 7 feet diameter by 14 feet high, and 100-pounds pressure, have been fitted, together with a "Contraflo" auxiliary condenser, and a small Morison evaporator. Either boiler is capable of doing the work, and only one is under steam at sea. Arrangements have been made to drive the steering gear by compressed air from the main-engine compressor, and this has been found so satisfactory that air is always used when at sea.

The *Saltburn* engines have cylinders 20½ inches, 33 inches and 54 inches by 36 inches running at 62 r.p.m., with two single-ended boilers, 13 feet in diameter by 10 feet 6 inches long, and 180 pounds working pressure, the engines being fitted with an ordinary surface condenser, Morison feed heater and evaporator. There is the usual outfit of steam winches, steering gear, &c., and one auxiliary boiler, 7 feet in diameter by 14 feet high, and 100 pounds pressure.

### COMPARATIVE TRIALS OF THE "SALTBURN" AND "EAVESTONE."

The working results, based on five voyages of each ship, showed that the *Eavestone* had an average speed of 8.75 knots with an oil consumption of 3.64 tons per day, and a coal consumption of 13 cwt. per day in the auxiliary boilers; whilst the speed of the *Saltburn* was 8.4 knots with a consumption of 12.1 tons. The deadweight capacity of the Diesel-engine ship is only 20 tons more than that of the other vessel, which is accounted for by the extra weight which it was thought wise to put into the first engine of this type, and by the additional auxiliary boiler. It is anticipated that this weight can be considerably reduced, especially if the auxiliary boilers are dispensed with and the auxiliary machinery driven by electricity or compressed air. The chief saving in weight, however, is in the smaller quantity of fuel to be carried for the Diesel engine. Thus, if both ships were loaded for a 30 days' voyage to be made at the same speed the *Saltburn* would have to ship 405 tons of coal, whilst the *Eavestone* would take only 110 tons of oil fuel and 20 tons of coal for the auxiliary boiler, a total of 130 tons, or 275 tons less than the steamship, and consequently the Diesel ship could carry that extra weight of paying cargo. The cost of the crew is in favor of the Diesel ship, as no firemen are required. On the other hand, the cost of the Diesel engine is more than that of the steam engine. The foregoing particulars appeared in an article in "Cassier's Magazine," by Mr. T. Westgarth.

### ADVANTAGES AND DISADVANTAGES OF THE DIESEL ENGINE.

In the same journal Engineer Lieutenant W. P. Sillence, R. N., discusses the advantages and disadvantages of the marine Diesel engine, and expresses his views as to the probable lines of development. He anticipates the extended use of this means of propulsion for tank steamers employed in the transport of oil fuel in bulk, and for cargo vessels in general when supplies of oil fuel can be obtained at a sufficiently low price. On the subject of price he points out that, taking triple-expansion steam engines using about 1½ to 1¾ pounds of coal per I.H.P. per hour, and Diesel engines using, say, ½ pound of oil per B.H.P. per hour, the cost of fuel per hour for the same B.H.P. would be less for the Diesel engine even if the oil is 3½ times to 4 times the price of coal. The relative prices are much more favorable than this in many parts of the world, and in many steam engines the consumption figures given above should be doubled. On the other hand, he considers that the early adoption of Diesel propelling engines for the largest ocean passenger liners is unlikely, as much preliminary experimental work will be necessary, and he adds that "a serious question, viz: continuity of supply and stability of prices of suitable fuel, remains to be answered." He is, however, emphatically in favor of the employment of Diesel engines for auxiliary purposes in large mercantile vessels, the auxiliaries to be electrically driven, and the current generated by Diesel engines placed in a central dynamo room. On the subject of Diesel engines for naval service he points out that naval vessels are not maintained in commission in order to earn money, and that Diesel engines, if they could be installed, would secure for such ships a large reduction in the amount of fuel used at any speed, and a corresponding increased radius of action for the same total weight of fuel carried. Diesel engines, he adds, for use at cruising speeds will maintain a high economy at such speeds, and Diesel engines for auxiliary purposes will enable the ships to use electric light and power in harbor with a minimum of attendance, and with considerable economy, due to their independence of boilers. For reasons which he gives, he states that the propulsion at full speed of torpedo-boat destroyers by means of Diesel engines is not practicable, but he regards the Diesel engine as an ideal source of power for submarines.

### THE FUTURE OF OIL FUEL.

It is, therefore, evident that the future of oil fuel depends not only upon the extent to which it is reasonable to assume that the output can be increased, but largely upon the manner in which the fuel is used, for in the light of our present knowledge it would obviously be wrong to suggest that the supply can ever become so abundant as to give consumers in general a free choice in substituting oil for coal as a source of power for industrial purposes, especially if the oil is not used in the most economical manner. The petroleum industry is one of great magnitude and importance; it never has been in a more vigorous condition than it is in today; it is unquestionably capable of much further expansion, and the use of fuel will undoubtedly largely increase; but although it is impossible to say what further stores of petroleum remain to be discovered, it is evident that it will require a very considerable expenditure of capital to greatly increase the average rate of expansion of the petroleum industry, especially as some of the older oilfields are showing signs of exhaustion.

What may be the position even in the immediate future it is impossible to predict with any approach to precision, owing to the uncertainty of the factors. There will doubtless be a continuous and large increase in the

requirements of the navies of the world, and as this demand is not primarily governed by price, it may be described as irresistibly preferential in character, taking precedence therefore over all industrial needs. Further rapid progress may also be expected in the use of the more volatile products in road motor vehicles, and in some countries favorably situated in respect of supplies there will doubtless be a greatly extended employment of oil fuel on railways. In respect, however, to the general substitution of oil for coal as a source of power in industrial establishments on land, it does not seem reasonable to anticipate more than the adoption of the former fuel to the limited extent commensurate with a gradual increase in the output, and with the surplus available after other demands have been satisfied. There will, therefore, be competition amongst consumers to obtain supplies, and as regards stationary sources of power it is improbable that, excluding exceptional cases, the oil will be used otherwise than in an engine of the Diesel or semi-Diesel type. The latter type of engine is receiving special attention in this country, whereas on the Continent it has been neglected for the Diesel engine, and it should be remembered that the former is somewhat less economical in consumption and more exacting as to quality of fuel than the latter, the consumption of oil per B.H.P. hour being about 0.6 pound.

Except as regards the belligerent navies of the world, in respect of which the dominant factor is efficiency, it is evident that any increased use of oil as a source of power must depend upon the price at which the fuel can be obtained. In the use of oil for steam raising, and in other industrial operations, the consumer can afford to pay about twice as much for oil as he pays for coal, in view of the higher thermal efficiency of oil and other advantages attaching to its use, but if the market value of oil fuel becomes established on what may be termed the Diesel-engine basis, it is only those who occupy exceptionally favorable geographical positions in respect to sources of supply who could hope to employ that description of fuel in steam raising.—"Shipbuilding and Shipping Record."

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## COMPARATIVE STRENGTHS OF NAVAL POWERS.

The usual method of comparing the strengths of fighting efficiency of the various naval powers, by simply comparing the number of capital ships, only gives a very slight representation of the actual comparative strengths of two navies. If, for example, a certain navy had two super-Dreadnoughts each mounting twelve 12-inch guns, whereas another navy possessed two super-Dreadnoughts each mounting eight 12-inch guns, by the usual method of comparison these navies would be considered of equal strength, but if reckoned by gun power, the former has twenty-four 12-inch guns as compared with the latter's sixteen 12-inch, the former, therefore, having an extra fighting efficiency of eight 12-inch guns, which may be considered of almost equal strength to two battleships of the pre-Dreadnought type. It is, therefore, obvious that the method giving the fairest comparison is that by which the main armament is the first consideration, the number and size of the vessels being a secondary consideration. The following table, compiled from the standard works of reference, shows a comparison between the British and German navies, these being considered the premier Naval Powers of the world. Only battleships constructed within the last twenty years may be considered as still efficient fighting units.

*Great Britain.*

| Type or class. | Number of ships. | Main armament. 13.5 in. | 12 in. | 9.2 in. |
|---|---|---|---|---|
| Battleships— | | | | |
| *Majestic* | 9 | ... | 36 | ... |
| *Canopus* | 6 | ... | 24 | ... |
| *Formidable* | 8 | ... | 32 | ... |
| *Duncan* | 5 | ... | 20 | ... |
| *Swiftsure* | 2 | ... | ... | 8 (10″) |
| *King Edward VII* | 8 | ... | 32 | 32 |
| *Dreadnought* | 1 | ... | 10 | ... |
| *Lord Nelson* | 2 | ... | 8 | 20 |
| *Bellerophon* | 3 | ... | 30 | ... |
| *St. Vincent* | 3 | ... | 30 | ... |
| *Neptune* | 1 | ... | 10 | ... |
| *Colossus* | 2 | ... | 20 | ... |
| *Orion* | 4 | 40 | ... | ... |
| *King George V* | 4 | 40 | ... | ... |
| *Iron Duke* | 4 | 40 | ... | ... |
| Battle cruisers— | | | | |
| *Invincible* | 3 | ... | 24 | ... |
| *Indefatigable* | 1 | ... | 8 | ... |
| *Australia*, *New Zealand* | 2 | ... | 16 | ... |
| *Lion* | 2 | 16 | ... | ... |
| *Queen Mary* | 1 | 8 | ... | ... |
| *Tiger* | 1 | 8 | ... | ... |
| Armored cruisers— | | | | |
| *Cressy* | 6 | ... | ... | 12 |
| *Drake* | 4 | 4 | .. | 8 |
| *Black Prince*, *Warrior* | 6 | ... | ... | 36 |
| *Shannon* | 3 | ... | ... | 12 |
| | 91 | 152 | 300 | 128 |

*Germany.*

| | | 14 in. | 12–11 in. | 9.4 in. |
|---|---|---|---|---|
| Battleships— | | | | |
| *Hagen* | 3 | ... | ... | 9 |
| *Kaiser Fredrich* | 5 | ... | ... | 20 (9.2″) |
| *Wittlesbach* | 5 | ... | ... | 20 |
| *Braunschweig* | 5 | ... | 20 | ... |
| *Deutschland* | 5 | ... | 20 | ... |
| *Nassau* | 4 | ... | 48 | ... |
| *Ostfriesland* | 4 | ... | 48 | ... |
| *Kaiser* | 5 | ... | 50 | ... |
| *Ersatz Weissenburg* | 4 | 40 | ... | ... |
| Battle cruisers— | | | | |
| *Van der Tann* | 1 | ... | 8 | ... |
| *Moltke* | 2 | ... | 20 | ... |
| *Seydlitz* | 1 | ... | 10 | ... |
| Armored cruisers— | | | | |
| *Fürst Bismarck* | 1 | ... | ... | 4 |
| *Prinz Heinrich* | 1 | ... | ... | 2 |
| | 46 | 40 | 224 | 55 |

*Summary.*

| Power. | No. of capital ships. | 14 or 13.5 in. | 12 or 11 in. | 9.4 or 9.2 in. |
|---|---|---|---|---|
| Great Britain | 91 | 152 | 300 | 128 |
| Germany | 46 | 40 | 224 | 55 |
| Per cent. more | 97.8 | 280 | 33.9 | 132.7 |

NOTE.—Tables include all battleships, battle cruisers and armored cruisers.

From the above table it will be seen that Great Britain has nearly 100 per cent. more vessels, and the total main armament of these is 580 guns against 319 on the German ships, showing a superiority of numbers of 81.8 per cent.—"Shipbuilding and Shipping Record."

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## INCREASES IN PAY OF OFFICERS AND MARINE ENGINEERS.

In the "Shipbuilding and Shipping Record" of June 12 last we dealt with the wages of officers and seamen and gave, in the case of the former, a comparative table of the rates paid by twenty-four representative companies to first, second and third officers in 1907 and 1913, which showed that very substantial increases had been made varying from £1 to £3 per month according to grade. These advances had in most cases been made by shipowners on the representations of the officers' organizations—notably the Imperial Merchant Service Guild—whilst in other cases the companies dealt directly with their employees.

Thus, so far, anything in the nature of a strike was avoided; but, as we foretold at the time, this would certainly come about unless other owners responded to the demands of the officers; and this forecast has since been verified by the strikes, happily soon settled, that occurred in one or two lines. The result of these agitations has been a general revision of the scale of pay all round and more generous conditions of service, such as the granting of regular annual leave on full pay, improvement in accommodation, the establishment of superannuation funds where they did not already exist, and compulsory retirement of commanders so as to allow better opportunities of promotion.

### PASSENGER STEAMERS.

These conditions have now become general in the large passenger-carrying lines, and the results, so far as pay is concerned, may be summarized as follows:

*Present Rates of Pay for Officers.*

| | Chief. £ | Second. £ | Third. £ |
|---|---|---|---|
| First-class vessels | 17–25 | 14–18 | 11–16 |
| Second-class vessels | 14–18 | 12–13 | 10–11 |

A corresponding advance has been made in the case of juniors, whilst commanders' pay now ranges from £400 to £1,100.

### TRAMP STEAMERS.

In the case of tramp steamers, the rates of pay were settled by the General Shipowners' Society in March, 1913, on the following basis, which represents approximately £1 per month advance all round:

| | Chief. £ s. d. | Second. £ s. d. | Third. £ s. d. |
|---|---|---|---|
| Under 5,000 tons | 12 0 0 | 9 0 0 | 7 0 0 |
| 5,000 to 8,000 tons | 12 10 0 | 9 10 0 | 7 10 0 |
| Over 8,000 tons | 13 0 0 | 10 0 0 | 8 0 0 |

In some cases it is possible to obtain certificated officers as third mates at under the rates quoted, and as an inducement many owners are supplying bed and bedding.

In considering the increased burdens placed upon the shipowner, however, not only in the deck department, but in the engine room (to which we shall presently refer), it must not be forgotten that the demands of officers have been based upon the exceptional period of prosperity enjoyed by the industry since about the middle of 1911 and which lasted up to the first half of the past year. Since then there has been a steady decline in the freight market, and, according to some owners, the outlook for the future is bad. Moreover, the profits earned in recent years have scarcely compensated for the losses experienced in 1908, when something like 500,000 tons of shipping was laid up.

### THE ENGINE-ROOM STAFF.

To come now to the marine engineers: The conference which took place in the spring of 1912 between the Amalgamated Society of Engineers, the Marine Engineers' Association, the Steam-Engine Makers' Society, on the one hand, and tramp owners on the other, resulted in a regular basis or national scale being adopted for cargo steamers of the ordinary type. The societies mentioned represent the great bulk of engineers, and the owners practically the whole of the cargo-carrying trade, excepting liners (dealt with below), oil-tank steamers and weekly steamers.

The scale as agreed to on May 1, 1912, is as follows:

*National Scale for Engineers on Cargo Steamers.*

| | Chief. £ s. d. | Second. £ s. d. | Third. £ s. d. | Fourth. £ s. d. |
|---|---|---|---|---|
| Over 8,000 tons deadweight, Summer, F.B. | 18 0 0 | 13 0 0 | 9 0 0 | 7 0 0 |
| 5,000 to 8,000 tons deadweight | 17 0 0 | 12 10 0 | 8 10 0 | 7 0 0 |
| 3,500 to 5,000 tons deadweight | 16 10 0 | 12 0 0 | 8 0 0 | 6 10 0 |
| 2,000 to 3,500 tons deadweight | 16 0 0 | 11 10 0 | 7 10 0 | ... |
| Under 2,000 tons deadweight | 15 10 0 | 11 0 0 | 7 0 0 | ... |

Certain other arrangements were also arrived at. For instance, it was not insisted upon that the number of engineers it had been customary to carry should be increased.

Figures showing the relative increases over the old rate in vessels of over 4,000 tons are not available, but the following table, compiled from the schedules of wages now paid in cargo steamers of 1,000 to 4,000 tons at the principal ports, gives a fair idea of the working of the new scale. The figures are taken from the wages sheets of some 200 vessels:

*Comparative Table of Wages of Marine Engineers (National Scale) in Tramp Steamers of 1,000 to 4,000 Tons.*

*New Scale.*

| Port. | Average wages per month. 1st Eng. £ s. d. | 2d. £ s. d. | 3d. £ s. d. | Average net increase. 1st. £ s. d. | 2d. £ s. d. | 3d. £ s. d. |
|---|---|---|---|---|---|---|
| Bristol | 16 10 0 | 12 6 8 | 8 6 8 | 0 15 0 | 1 3 4 | 0 15 10 |
| Cardiff | 16 7 6 | 12 5 10 | 8 5 10 | 0 15 6 | 1 1 9 | 0 18 0 |
| Glasgow | 17 0 0 | 12 8 0 | 8 8 0 | 0 10 8 | 1 4 0 | 0 11 9 |
| Liverpool | 17 1 0 | 12 9 0 | 8 15 3 | 0 16 6 | 0 17 7 | 0 7 0 |
| London | 16 11 6 | 12 2 6 | 8 3 6 | 0 12 7 | 0 18 6 | 0 9 6 |
| Sunderland | 16 15 0 | 12 5 0 | 8 5 0 | 0 7 6 | 0 17 6 | 0 15 0 |
| Newcastle-on-Tyne | 16 14 0 | 12 3 10 | 8 3 10 | 0 18 3 | 1 3 4 | 0 15 4 |

ENGINEERS ON LINERS.

With regard to liners, the rates vary considerably, according to the class of steamer, but the following may be taken as fairly representative of the maximum pay after, say, five years' service:

| | Chief. | Second. | Third. | Fourth. | Fifth. |
|---|---|---|---|---|---|
| | £ s. d | £ s. d. | £ s. d. | £ s. d. | £ |
| First-class liners | 29 0 0 | 18 0 0 | 14 0 0 | 11 10 0 | 9-10 |
| Second-class liners | 25 0 0 | 15 0 0 | 12 10 0 | 9 0 0 | 8-9 |

In arriving at the above figures, bonuses and "run money" have not been taken into account, but in many vessels these combined would add anything from £30 to £150 per annum to the salary of the chief engineer.

To what extent the improved conditions will affect the executive and personnel of the service at large remains to be seen. So far as can be gathered it has not as yet helped to swell the ranks of youths willing to adopt the profession, mainly, we believe, because the general public are not sufficiently enlightened, but it is to be hoped that in a few years it will be recognized as offering inducements comparable with other professions which, so far as initial outlay for education and training is concerned, are certainly more costly.

The following list gives a few representative companies that have granted increased pay during the past year:

P. & O., Orient, Royal Mail, White Star, Cunard, Pacific Steam Navigation, British India, Atlantic Transport, New Zealand Shipping Company, Shaw, Savill & Albion Line, Allan Line, Union-Castle, Asiatic Steam, Bucknall's, Clan Line, Elder-Dempster, General Steam, Indra Line, Tyser Line, Wilson Line, Booth Line, Houlder Brothers, T. Dunlop & Sons, Furness, Withy & Co., Manchester Liners, Moss Steamship Company, Canadian Pacific Railway Company.—"Shipbuilding and Shipping Record."

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## THE HABIT OF OBSERVATION.

The first lessons man received on earth were the results of observation. Primeval man, in observing the flashes of lightning, became aware of its power and its destructiveness. Probably his first idea of making fire was the result of observing the flying sparks from rocks as they slid over the surfaces of other rocks. Perceiving this result of friction he developed a means of making fire.

From the observation of physical and scientific phenomena man found insight into its causes,—thus our scientific laws have been deduced. "The master observer of modern time was the sagacious and versatile Franklin." His prophetic and comprehensive insight into the ultimate cause of natural phenomena placed before us the nature of the electric fluid, the properties of matter, and the theory of earthquakes. It was only an inquiring mind, seeing through critical eyes which the minutest detail could not escape, that enabled Pasteur, another paramount observer, to revolutionize the biological theories of his day, and found the modern science of Bacteriology.

To the student we would call attention to the potentcy of observation. Students too often forfeit the power of a clear and independent visual observation. They loose the power of individual and accurate judgment by relying upon others to do their thinking. The engineering student is inclined to become the slave of the machine and treadmill. As a result of his specialization his observation is more or less confined to that of a technical nature. Yet few engineering students cultivate the habit of

independent observation, *even of a special character.* Too frequently reports on a mechanical experiment or test of a machine are returned to the student because he failed to observe an important action or phenomenon. He may have seen these things, but only in a *blind stare.* "The Reagents,—apprehension, comparison, judgment, reflection, do not act; the image is not fixed upon the consciousness and filed away by the memory for future references, and the man's intelligence is so much the weaker." Too many students do not train their minds to retain the important points,—results of their observations. Why can one man, with just a fleeting glance enumerate those things that make up a heterogeneous mass of substance? It is the result of concentration and a mind cultivated for independent observation.

The importance of cultivating this habit of thorough and independent observation cannot be emphasized too vigorously. Although many advocate that this should be the work of the primary and grammar schools, it nevertheless is a habit that the college man *can* cultivate,—if he would.

Beware of specialized observation. The habit of a critical, independent, yet broad observation, is an education in itself. And aside from this, when we have forgotten most of the accumulated facts which we are all too prone to collectively call an education, the habit of observing remains with us. Franklin does not stand the master observer in an engineering and scientific capacity alone, but in the broader viewpoint,—including nature, art and humanity.

What enjoyment can be obtained in nature's beauty, what pleasure in the varieties of the human make-up, what an education in art. Consider what opportunities elude our grasp, what a wealth of delightful, intelligent perception, what a keen insight into human nature is veiled from us, through the lack of habits of broad observation and refined organs of sense.

Many engineering students will disagree with the esthetic viewpoint. Why? The rut is clear and seems to be a natural fall,—specialization. That cause will be all too prominent in practice, and by falling into it now is depriving yourself of your main object,—*a liberal education.* It is for this reason that the habit of a broad observation is to be cultivated. The successful engineer of today is putting into practice the results of a liberal, yet individual observation,—*individual judgment.* He not only deals with the forces of matter and the laws of gravity, but must harness the forces of human nature. For a diversity in his work he seeks the more esthetic things. Critical observation means intelligent application and greater efficiency. Observe!—"The Sibley Journal of Engineering."

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## AUXILIARIES.

### THE "STANDARD" ALTERNATING-CURRENT MARINE STEERING GEAR.

#### SYNCHRONOUS MOTION, FOLLOW-UP SYSTEM.

The "Standard" polyphase alternating-current electrical steering gear is one of the most interesting results of the original research work carried out by Mr. W. P. Durtnall in connection with his high-frequency-current lighting at the Cannon Street Hotel in 1895, together with improvements recently effected with his co-inventor, Mr. Algernon H. Binyon, the late technical adviser to the R. A. C.

The "Standard" electrical gear is interesting from the fact that there is no waste due to resistances in the main circuits, and the absence of commutators on the main motor makes for reliability, especially when bear-

ing in mind the enormous vibration that is always present at the stern of a screw-propelled vessel of any kind. It is an invention which is the result of the study of many kinds of steering gears, not only those operated by steam, but also those operated by electricity or hydraulic gear. The new device represents an electrical differential variable gear, and controlled in such a manner that synchronous motion takes place between the control lever on the navigating bridge and the rudder, the rudder being compelled to "follow up" the movements of the control lever, and to whatever degree of helm is required, at any time. Further, the rudder is brought to any degree of helm, and is there automatically electro-magnetically locked in any particular position at the will of the navigating officer, who is also able by means of the electrical helm indicator provided to watch the movement of the rudder in any port or starboard direction.

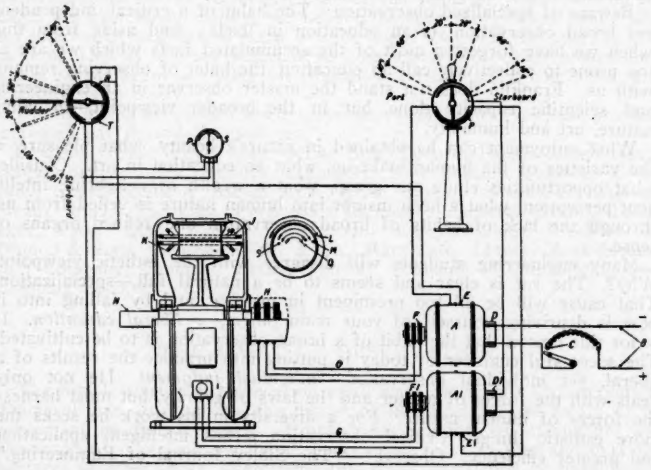


DIAGRAM SHOWING WIRING AND OTHER ARRANGEMENTS OF THE "STANDARD" ALTERNATING-CURRENT ELECTRICAL MARINE STEERING GEAR.

The plant shown in the drawing here produced represents the system designed for working from the ordinary direct current, say, 200 volts electric-lighting buss-bars in the engine room; but the system is also designed to be run with separate steam engines or turbines, or other prime movers, such as internal-combustion engines, as may be thought desirable. In the engine room are installed two rotary converters, marked A and A1. Current is drawn at, say, 200 volts direct current from the ship's mains M, through the starting switch marked C, this current going to the two commutators marked D and D1. The field current is also sent to the field coils after passing through the bridge and rudder controls, and enters the field coils at terminals marked E and E1. In this case we will assume that the rotary converters are wound for four poles, and when the bridge control lever is in the position shown at O, both converters run at

a revolution speed of 1,000 r.p.m. Three-phase alternating current comes off the slip rings shown at F and F1 at a voltage of about 115 and at a frequency of 4,000 alternations per minute (33⅓ cycles).

The three-phase alternating current is led to the steering motor through the conductors shown at G, and these conductors are coupled to the motor in the following fashion: The motor is wound for (in this case) six poles in both stator and rotor, and each motor winding is also for the same voltage with equal number of conductors per slot, the current going into the rotor and stator windings respectively through the slip rings H, and the stator terminals I. The stator windings are shown at J, and the rotor winding at K. The current is so connected that the resulting magnetic flux in the stator iron revolves in the direction shown at L, (on the small end-view diagram), whilst the magnetic flux in the iron of the rotor revolves in the same direction and is shown at M; whilst the lever O on the bridge is in the position shown, the rotary converters run at 1,000 r.p.m., but the rotor is electro-magnetically locked, and consequently the rudder is locked in line with the keel of the ship.

The power shaft of the motor is shown at N, which is coupled to the worm or other gear that delivers the mechanical power produced in either direction by the motor to the rudder head and quadrant. A Wheatstone bridge balanced shunt resistance is shown in the bridge control at P, and the other Wheatstone bridge resistance is shown at the rudder head at R and T, the positions shown for all controls being that when the ship would be traveling in a straight line ahead.

The operating conditions are as follows: Assume that the navigating officer wishes to go to 20 degrees a-starboard, he pulls over the lever O, on the navigating-bridge pedestal, to mark 20 degrees. By tracing the connections through it will be seen that by so doing he places more resistance in the field of the converter A1, and also reduces the resistance in the field of the converter A, which immediately allows the A converter to slow in speed and the A1 converter to rise in speed, so that there is an immediate difference in the frequency starting from zero to a difference in the rotor frequency of 1,304 alternations per minute, by the fact that the A converter falls in speed to 840 r.p.m. and the A1 converter rises in speed to 1,166 r.p.m.; the rotor then turns in the direction marked in the small diagram at Q, being six pole runs at 217 r.p.m. As the rudder comes round to the 20 degrees to starboard position the resistance in the shunt of the A1 converter is gradually cut out, and further resistance is inserted in the field of the A converter, so that when the rudder gets round to the 20 degrees a-starboard position the converters are then running at the same speed again, and the rotor and the rudder is electro-magneticaly locked in the position. By bringing the control handle back to the central position on the bridge pedestal resistance is let into the field of the A converter and cut out of the field of the A1 converter; there is an immediate difference in the frequency from zero in the rotor and stator circuits of the motor, the stator receiving a frequency of 4,664 alternations per minute, and the rotor only 3,360 alternations per minute, so that the rotor turns in the direction marked in the small diagram at S; the rudder is so brought back again to the central position. As this is being done the resistance in both fields of the converters is balanced again, so that the rotor and the rudder is again locked in the position as controlled at the bridge-control pedestal and under the direct handling of the navigating officer.

Should a quick turn to port be required, the officer can pull the lever over to port to the extreme position of, say, 60 degrees and immediately the converter A runs up to a speed of 1,500 r.p.m., and the converter A1 drops in speed to 500 r.p.m., the six-pole motor driving the rudder round to the 60 degrees to port position, part of the time running at a speed of 666 r.p.m., but getting slower as the rudder comes round by reason that

it will be seen that the rudder-head Wheatstone balance resistance is being gradually put into the field of the A converter and cut out of the A1 converter, so that when the rudder reaches the 60 degrees position the converters come at the same speed, and there is no motion at the rotor of the motor because both magnetic fluxes in stator and rotor are revolving at the same revolution speed again.

The same action taking place to the a-starboard direction as may be desired, the extreme run allowed in the Wheatstone balance resistance is for 70 degrees. This is arranged in this manner so that should the bridge-control lever be left at, say, the extreme of 60 degrees, there cannot be any over-running against stops at the rudder quadrant, because as soon as the rudder gets to the 60 degrees position the motor automatically stops for the reason given above; so the last 10 degrees are left to play on, the 60 degrees being sufficient for all practical purposes of navigation. The above type of electrical steering gear is simple both in construction and reliable in operation, with great efficiency in working. It is being placed on the market of the world by the "Standard" Engineering Association, of 76, Gladsmuir Road, Whitehall Park, Highgate, London, N., and it is stated to represent the cheapest electrical steering gear on the market at the present time.

The helm indicator consists of a double-coil voltmeter, arranged so that the voltage between the ends of the rudder-head resistance is registered and calibrated on the dial in rudder degrees to port or starboard, according to the position taken up during its course.

In the case of very large ships the rotary converters are replaced by engine or turbine alternators, and the steam stop valves are under control by means of solenoids controlled in like manner to that described above, so that the engine or turbine runs at different revolution speeds in the same direction of speed at all times. In the case of ships fitted with internal-combustion engines for main propulsion, this steering gear presents a means by which the usual steam plant required at sea to operate the steering gear can be done away with and thereby, it is claimed, greater steering efficiency be maintained, and at lower first and running costs, while the work of the navigation officer is reduced insomuch as the largest ship afloat could probably be steered by one's little finger.—"Shipbuilding and Shipping Record."

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## THE GYROSCOPE AND GYROSCOPIC ACTION IN ENGINEERING PRACTICE.

At the Institute of Marine Engineers, on 20th October, a paper on "The Gyroscope and Gyroscopic Action in Engineering Practice" was read by Mr. Thos. R. Thomas. Mr. Alexander Boyle (vice-president) occupied the chair. In the course of his paper, Mr. Thomas said every rotating wheel was not necessarily a gyroscope. Gyroscopic action and intensity depended upon (1) the moment of inertia of the rotating wheel about its axis—in other words, it depended upon the diameter and weight; (2) its freedom to incline its axis of rotation in one or more directions; (3) the velocity with which the axis of the wheel was inclined; (4) the velocity with which the wheel was rotating.

Many years ago the gyroscope was used for determining the angle of rolling in ships, advantage being taken of the fact that a gyroscope resists any movement from the plane in which it is rotating. The ordinary pendulum would be of very little use for this purpose when accurate meas-

urements were required. It had also been applied for the prevention of rolling in ships, and a description was given of its application in this direction. By means of a brake the inclining of the wheel was restrained, consequently its action was spread over a longer time, but the intensity of the righting force was diminished. This would make it clear that the gyroscope would be much more effective in a small quick-rolling vessel than in a large one with slow roll.

The gyroscope was also used in the Whitehead torpedo for automatic steering, as only by such means could the torpedo be made to keep a mean straight course.

It had also been introduced successfully on board ship as a compass, and it had been applied to aeroplanes for maintaining stability by automatically adjusting the balancing planes, but it did not appear to have much success in this direction up to the present.

The rolling of a vessel, the author stated, produced no gyroscopic action in a wheel fitted on the screw shaft, as its axis was always parallel to the axis about which the ship rotated.

The author dealt in detail with the principle and action of the gyroscope in various departments of engineering.

Mr. J. G. Hawthorn, in opening the discussion which ensued, described the construction of the *Bessemer,* and stated that the gyroscope was applied to control the hydraulic ram which operated the swinging saloon on that vessel. He also referred to a vessel on which there was a noticeable absence of pitching. The engines were fitted with a heavy flywheel, and he considered that this might have had some effect upon the vessel in respect of its movements in a heavy sea way.

Mr. E. Kilburn Scott cited a case where gyroscopic action caused the breakage of the shafting in a ship-lighting installation when the dynamo was fixed athwartships. He had been deeply interested in the application of the gyroscope in connection with the Brennan monorail, but there were various difficulties in the way of its successful operation. He considered that the most successful method in which it could be applied was in the direction of ships' compasses, especially for battleships. The two gyroscopic compasses which had given the most successful results were the Anschultz and the Sperry. The peripheral speed adopted for both compasses was the same as that used by Brennan, viz: 33,000. For these compasses special motor generators had to be constructed, with a periodicity of 330 per second.

Mr. James Shanks said the author had demonstrated that a wheel fixed upon a propeller shaft could have no effect upon the movement of the ship and, therefore, the flywheel could not affect the pitching. It was evident also that it was impracticable to apply the gyroscope on a large scale to prevent the rolling of ships. The power of the gyroscope in this direction was inversely proportional to the period of rolling, and in large vessels with a long period of, say, 14 seconds, it would mean that the gyroscope would have to be of such dimensions as to make its use prohibitive.

Mr. F. O. Beckett instanced a case of a vessel the engines of which were fitted with two flywheels, and the ship had little or no rolling or pitching movement, which fact he attributed to the gyroscopic action of the wheels. In reference to the use of the gyroscope as a compass, he asked whether the indication would allow for the deviation of the earth's axis.

In reply to the last speaker, Mr. Thomas said the axis of the gyroscope would always assume a position parallel to the earth's axis. He demonstrated that a wheel rotating on a fixed axis offered no resistance, and that the flywheel, therefore, could not affect the rolling of the ship.—"The Steamship."

## THE PRESENT VALUE OF ARMOR.

Among all the innumerable estimates made in times of peace of the probabilities of what may happen under conditions of war, it would appear that greater difficulty is involved in arriving at the most suitable distribution and thickness of armor for a given vessel than at the required proportion of any of the other main essentials in the design. This difficulty, it must be frankly admitted, has become one of considerably increased moment in recent years, and, in fact, at the present time probably no other of the components exerts a more important influence on the design than do the protective arrangements of modern warships. It will readily be admitted that the conditions have altered more widely and rapidly in the last decade than during any other period, but it is often imperfectly understood that they have so entirely altered in that period that any deductions now made from even the illuminating actions in the Russo-Japanese War are of comparatively little value. The size of ships and the number of heavy-gun positions in each has been more than doubled since the battle of Tsushima, and the 40-caliber 12-inch gun which was then the most powerful weapon in the two fleets, has now been superseded by the 15-inch gun of 45-calibers, which possesses double the muzzle energy and a proportionately much greater superiority at modern battle ranges. The resisting power of armor has certainly made some advance in recent years, though not to any great extent—certainly not, for instance, by any means comparable with that which occurred when the Krupp process was substituted for the Harvey system of manufacture—and in its present condition the attack must be frankly admitted to be superior to the defence, although there is naturally a certain body of opinion, which prefers to qualify this view on the grounds that increased subsidiary protection is being more generally afforded.

No adequate method, however, of meeting the destructive power—not necessarily the purely trial ground perforation of the modern big gun so much as its smashing effect—seems possible except by considerably increasing the thickness and area of the armor plate. This tendency is now a marked feature of all contemporary battleship designs, though possibly more specially exaggerated in the case of the recent United States vessels, and it is moreover generally associated with a considerable extension of side protection, not only in length, but in both an upwards and downwards direction from the water line. Quite apart from the military value of armor bearing any proportion to its price, which alone happens to represent about a quarter of the complete cost of a modern ship, the question as to whether it really is worth while putting as much as, or more than, 6,000 tons of protective material into a ship whose normal displacement is about 27,000 tons, is one to which too much consideration cannot be given in view of the truly enormous influence that this proportion of the total displacement has on the design of the complete ship. In recent cases, the ratio of armor weight to normal displacement has ranged from 20 per cent. in the large armored battle cruisers to 30 and to even nearly 35 per cent. in the case of some recent battleships, and it cannot be denied that in view of the fact that the corresponding percentage weight devoted to armament is only two-thirds of that devoted to armor, the proportionment of armor recently allotted would seem to be out of all proportion to the effectiveness of the protection secured in view of the relatively greater advance of gun power in the last decade.

For some time past there has been an increasing tendency on the part of many naval officers to argue against the idea of increasing belt and turret armor thicknesses; on the other hand, the demand is still advanced by some for putting the auxiliary or anti-torpedo boat weapons behind armor. The United States Navy Department, which has during the last seven years pursued a very consistent policy in battleship design, did put

its smaller guns behind armor in the *Delaware, Utah, Arkansas* and *New York* types, but in its 1911 ships, *Oklahoma* and *Nevada,* and in the *Pennsylvania* ordered this year, these weapons are placed in an entirely unprotected battery, while the heavy armor on belt and turrets has been very considerably thickened, the result being to produce an "all or nothing" design which is in direct contradiction to the general policy observed in nearly all European designs, which consists in carrying thinner belt and turret armor and armoring the 6-inch batteries. On the surface, the *Oklahoma's* protection would appear to have been achieved at the expense of far too great a proportion of the displacement. It is especially curious that during the period when the United States were protecting their auxiliary weapons the British Admiralty left its entirely exposed, but simultaneously with the armoring of the 6-inch weapons of the *Iron Duke* class, the Americans abandoned protection entirely for anything except big guns and belt, whilst protecting the latter on a vastly increased scale.

In view of our lack of modern war experience—and notwithstanding Tsushima or experimental firing trials at the *Hero, Deutschland, San Marcos* and *Empress of India*—the only method at present of correctly appraising the value of armor is based on a selection of the most important of the most probable risks that the permutations and combinations of modern battle conditions would seem likely to provide, and to guard against them with what is judged to be the minimum effective thickness of armor. Obviously, while probable risks can be multiplied, and the most important gaged with a fair degree of accuracy, there is an element of luck in the consequences of a hit which, history seems to show, should have a special margin of its own. And if one is to accept any theory of lucky hits and be willing to deduce even this much from past experience, why not go a little further? To do so is to embark almost at once on a sea of uncertainty due to such varying conditions that it is often only possible to accept both deduced and negative proofs rather than the apparent positive information conveyed. Armor saved the American Monitors again and again in the Civil War; one is tempted to wonder what its value can have been forty years later in the Sea of Japan. True, at Tsushima armored ships were sunk by gun fire, though with relatively little damage to belt armor, but the gun fire is now known to have been accurate only to a very small extent, and one may consequently be inclined to deduce that if the accuracy had been of a high order the existing armor would have been almost useless in spite of the fact that it was, according to all ideas current ten or fifteen years ago, of ample extent, thickness and distribution. Again, even at Tsushima, it was by no means the case that heavy armor either saved or failed to save the Russian fleet. The one modern battleship that survived, the *Orel*—now the Japanese *Iwami*—was forced to surrender by gun fire without her heavy armor being pierced at all; her upper works and superstructure, however, were a complete wreck. Japanese armor was not sufficiently often hit hard enough to teach us much.

The various experiments that have been carried out, culminating a few weeks ago with those on the *Empress of India,* have all yielded information of varying degrees of value, all of which almost invariably needs a considerable amount of theoretical correction for such facts as difference in age between armor and projectile, that accurate hitting can be arranged, that the target ship has not, so far, ever been put in a properly "cleared for action" condition—it is obviously impossible to do so—and, again, that invariably the target ship was designed to withstand war conditions totally different from those occurring on the firing trial. The experiments carried out in 1900 on H. M. S. *Belleisle,* against which H. M. S. *Majestic* fired no less than eight 12-inch common shell, eight A. P. shot, two hundred 6-inch shell, half ordinary and half lyddite, and

about four hundred 12-pounder shell, were fully described in "The Engineer" at the time. The average range was only 1,500 yards, and the average number of hits secured only about 33 per cent. The *Belleisle* was sunk by a 12-inch shell striking the water line and making a large hole, much as was the *Empress of India* more recently. In spite of all woodwork and inflammable material being left in the ship, the *Belleisle* was not set on fire. This, and the enormous difference in the result of the burst of a lyddite shell as compared with a common one, were among the principal facts elicited, but while the latter had already been proved in the South African war the former can only be set down to accident, for the ship was struck altogether by five 12-inch shells, seventy-five 6-inch, and about three hundred and fifty others, a quantity that one would have imagined would have done infinitely more damage. Extraordinary as it may seem, only two 12-inch and one 6-inch shell got through the armor. The net result of the experiments was said at the time to demonstrate very clearly the value of armor protection against shell fire, and that the unarmored parts of a ship were untenable under the fire of quick-firing guns. And now for the application of the corrections necessary. The first is that of the range, which was only 1,500 yards; at this distance the hail of 3 and 12-pounders on an old iron ship would be effective, but less than four years later, in the Sea of Japan, the ships were hardly ever within two or three times this range of one another. Again, the first couple of 12-inch shell ought to have completed the *Belleisle's* destruction in two minutes—in which event what would the verdict on armor have been? The necessity for armor to keep out shells was proved nearly sixty years ago at Kinburn and Sinope, but how are the conditions then, in the American Civil War, in the Spanish-American or Chinese-Japanese War, to be compared either with Tsushima or today? Admiral Custance, in his essay on "The Ship of the Line in Battle," has shown how in almost all cases armored ships have been beaten before they have been sunk, and even beaten without being sunk or without their armor being perforated, and even at Tsushima, from which battle it was expected to have drawn many more conclusions than were eventually found, the conditions were such as really only to corroborate certain theories without adding much to our knowledge of questions concerning the relative values of material.

If the question as to what is the present value of armor is to be honestly answered, it only can be in the light of knowledge of modern gunnery. As stated above, the attack for the moment is out of all proportion immeasurably superior to the capabilities of the resistance of the present systems of armoring ships. We know the havoc wrought on board the Japanese battleship *Mikasa* by an old-pattern powder-loaded shell of 12-inch diameter that penetrated the belt on the starboard bow just where it commenced to taper off from 9 inches to 4 inches, and it hardly involved much prophetic risks to guess beforehand what would occur to the *Empress of India* if a big modern shell of double the power got home near her water line. What, however, mostly concerns the naval designer of the present day is what is to be the main-belt and turret thickness under such circumstances. Are we to keep out shell but not shot? Even shell can penetrate fairly heavy armor. Is the modern belt to be 10 inches, 12 inches, or 14 inches thick? The difference between the two latter for a belt 450 feet long and 16 feet deep in no less than 500 tons. Roughly speaking, the saving on five turrets for a corresponding reduction of 2 inches would also be nearly as much. The influence of this on the displacement is very great and would probably exceed a total of over 2,000 tons, quite apart from its influence on stability.

Now it must be remembered that, apart from any increase of caliber, muzzle energy, and bursting charge of shells, the accuracy of modern ordnance has been increased in the last seven years enormously, the num-

ber of heavy guns carried has grown from four to ten, and the rate of fire per gun has improved by 50 per cent. That is to say, that ships with a typical modern broadside of ten guns—making the reasonable assumption that hits per gun per minute are at least double of what they were till only recently—are able to make no less than seven and a half times the number of hits in a given period of time, and if 13.5-inch shell are contrasted with the 12-inch of 1905, can, in that time, theoretically execute fifteen times the havoc possible at that period. The resistance of armor has not increased in anything like that proportion, but even if it be granted a fair measure, it will be seen that a modern ship has a potential value for destruction ten or twelve times what it was eight years ago, and this figure is enormously augmented when the greater ranges now attainable are also brought into consideration. What the value can be of the 8-inch and 9-inch side armor, or even of the narrow 11-inch and 12-inch waterline belts of all the British *Dreadnoughi* class is hard to determine in face of figures like this. The real fact is that, though the armor is penetrable, it is only penetrable by projectiles of a less dangerous nature than would be employed if the armor were thinner or non-existent. That is to say, one is forced to attack the ship with a shell having a smaller bursting charge in proportion to its weight. Hence, penetration is assured by directing a group of shells on to one spot, but the individual effect is less than it would be if the shell were designed to attack thinner armor. Naturally, internal structural arrangements have also been far more widely made to form subsidiary protection arrangements, but they are primarily intended as defences against mine or torpedo attack.

The experiment which would throw more light than any *Belleisle* or *Empress of India* experiments on this very vexed question would be to let two of the most modern vessels fire, at most, two rounds each at one another at 8,000 yards' range, at which distance it is at present supposed that a modern high-explosive armor-piercing projectile would be able to perforate 11 inches. That the cost would be high—enormous, in fact—is admitted, but the results would, we consider, go to show that the gun is still greatly underrated and that armor attached to a ship does not possess its apparent proof ground penetration by a very considerable margin. For it must not be forgotten that modern methods of fire control with range finders accurate to 2 feet per 1,000 yards, can place at four miles distance five simultaneous shell of the largest size in an area less than that occupied by a tennis court, and where a solitary shell might not succeed, the result of such a repeated blow is obvious. It would almost appear that the penetrability of modern armor under such conditions must be accepted, or that a radical change in design, directed towards a readjustment of balance of power between gun and armor, must make its appearance before long.—"The Engineer."

## THE PROTECTION OF CONDENSER TUBES.

The methods in use for the protection of condenser tubes are electrochemical in nature; they depend either on the use of some metal, such as zinc, iron, or aluminum or its alloys, which is electro-positive to brass, and which will consequently set up an electrical current when both are connected together and immersed in sea water, or on the use of a dynamo to pass the current to the brass. In principle the two methods are identical—namely, that of making brass which it is desired to protect a cathode, at which there will be a tendency for the liberation of hydrogen, while oxygen will tend to be set free at the anode. This anode may consist of any of the metals mentioned above. The idea is to concentrate corrosion on the anode, which is sacrificed to protect the cathode.

The metal zinc has been largely employed for anodes, both for the protection of boilers and condensers. In the case of the latter, it is placed in the water ends and is securely bolted to the tube plates. At first sight it would appear to be a more suitable metal than iron, since it is more electro-positive, but in practice this does not prove to be the case. There appear to be two reasons for this. The first is that the zinc gradually becomes covered with a fairly adherent layer of oxide, which itself is electro-negative to brass, and is a poor conductor of electricity. Thus the protective action of the zinc soon ceases, and might even be reversed but for the poor conducting properties of the oxide. The second reason is that the consumption of zinc is found to be great—far greater than that demanded by the requirements of electro-chemical protection. It seems probable that the zinc is subject to auto-corrosion due to the action of the zinc-oxychloride which adheres to the metal.

For the protection of condensers zinc has been largely replaced by iron, which is not only cheaper, but more efficacious. The iron is used generally in the form of slabs, which are placed in the water ends of the condenser, and are attached by bolts to the tube plates.

If any electro-chemical action is to occur, it is obviously necessary to insure that the connection between the iron slabs and the tube plate be thoroughly satisfactory. It is, of course, possible that the mere presence of iron slabs in the water-ends of a condenser might give rise to a certain amount of protection by depriving the water of a large proportion of its oxygen, which will go to form rust. If this be so, no special care need be taken to insure satisfactory electrical connections. To test this point, the following experiments were made:

Three pieces of 70:30 brass tube were taken, together with three pieces of wrought iron. On to one tube and one piece of iron lengths of copper wire were soldered. These were weighed separately, and then the copper wires were connected together, so as to insure good electrical contact, yet they could be separated again for weighing purposes. The tube and iron were then immersed in a beaker containing 1½ liters of sea water. One of the other pieces of brass tube was weighed, and suspended in another beaker at a depth of 2 inches from the surface. At a distance of ½ inch above it a weighed piece of wrought iron was arranged. Thus there was no electrical connection between them. The other length of tube and iron strip were similarly immersed in a third beaker through which a stream of air was kept bubbling. The air was introduced at the bottom of the beaker. All the beakers were then placed in a thermostat at 122 degrees F. The object of carrying out the experiments at this temperature instead of at the laboratory temperature, was to ascertain whether electro-chemical protection would prevent selective corrosion, which is particularly active at this temperature. At the end of seven and fourteen days all the pieces of metal were weighed. The tube which was in electrical contact with the iron was completely protected, and showed no loss of weight. The surface appeared quite bright and unoxidized, and no sign of pitting could be seen, although previous experiments show that it occurs normally in seven days at this temperature. On close examination the protected brass tube was found to be covered with a uniform thin transparent layer of some substance which could only be removed by scraping. The layer was of about the thickness of tissue paper, and proved to consist of calcium carbonate. The rôle of this layer evidently deserves careful consideration in any discussion on the mechanism of protection.

### ELECTRICAL CONTACT NECESSARY FOR PROTECTION.

Neither of the other tubes was completely protected, proving that electrical contact is necessary. The question then arises, how can electrical contact protect parts of the tubes remote from the contact, *i. e.*, why does

not all the current generated pass between the iron anode and the part of the tube near it?

It is difficult to see how iron slabs in the water ends can protect the middle of the condenser.

The same difficulty is met with in considering the mode of protection by a current generated by a dynamo, as in the Cumberland process. Of course, in this case a much larger current can be used, and it can be increased at will until it is effective, but the authors have been informed by Mr. Cumberland that in normal cases 2 ampères per 1,000 square feet of tube surface is sufficient for protection, and where such a small average current is used the amount conveyed by the most remote foot of tube must be small. It seems certain that some other factor must operate to assist protection.

To test the matter the following experiment was made:

In a glass vessel three 70:30 boiler tubes were placed at varying distances from an iron anode, the distances being ½ inch, 10 inches and 20 inches respectively, but as the last was twice the area of the first, it should receive one-twentieth the protective current, or what the middle of a condenser tube should receive.

At the end of three weeks the tubes were examined. Each was covered with a thin translucent layer of calcium carbonate. Under this the brass was found to be unattacked. A fourth test piece not connected was found greatly attacked, and without any protective coating. It becomes evident that when iron anodes are used for protection, that the greater part of the protective current passes at first by the shortest path and protects only the parts near the iron. These soon become covered with calcium carbonate, greatly increasing the local resistance, and the current passes to more remote parts, and covers them with the carbonate film, which thus gradually creeps along the tube. The extreme distance over which protection can be extended is conditioned by the relative resistance of the calcium-carbonate layer, and the increasing resistance of the more remote water circuits. The part played by this film in electro-chemical protection must be immensely important.

### HOW ELECTRO-CHEMICAL PROTECTION MAY BE DESTROYED.

Electro-chemical protection may cease to be effective through: Breakdown in the electrical connections; by mechanical injury to the calcium-carbonate layer by cleaning rods, etc.; by failure of the calcium carbonate to adhere to tubes already oxidized, corroded or dirty.

It becomes clear from this experimental work that dezincification, followed by pitting, is a phenomenon inherent in brass alloys of the type discussed.

The conditions for its appearance are the presence of certain kinds of water, of which natural sea water was one, and of a certain temperature. Dezincification may be prevented, or at least postponed, by four different methods: (1) By using a suitable water supply; (2) by using an alloy different from brass; (3) by adjustment of the temperature conditions when brass is used; (4) by electro-chemical protection.

In the mercantile marine (1) can never be employed, and it can rarely be used even in land installation. Moreover, there is as yet no precise information as to the kinds of water which will not give rise to dezincification. As regards (2), little information is at present available. An obvious suggestion is to lower the amount of zinc present in the alloy, and thus to diminish the tendency toward dezincification. As to how far, if at all, the elimination of zinc could be usefully carried, the authors cannot make any definite statement. It is, however, certain that a complete removal of zinc would give rise to serious trouble of another kind. Possibly there may be some alloy intermediate between 70:30 brass and pure

copper which would be more resistant to corrosion than either, though whether it could be satisfactorily drawn into tubes is another matter. Probably manufacturers have already experimented privately along these lines, and information on the subject would be welcomed.

A number of alloys which contain no zinc suggest themselves as suitable for the manufacture of condenser tubes. Phosphor-bronze has been proposed on a number of occasions. It can be drawn into tubes of good quality, and at first sight would appear to be an excellent material for the purpose. Under certain conditions, however, it appears to be subject to a peculiar and deeply penetrating form of complete corrosion, the cause of which has not yet been ascertained by the authors, and the phenomenon of "detinning" has also been observed. More work upon this alloy is desirable before it can be recommended for use. A copper-nickel alloy produced on a large scale in America, and known as "monel metal," has been used for condenser tubes. It appears to give satisfactory results as far as resistance to corrosion is concerned, but its high price has prevented its extended use. [This seems to contradict the Edison Illuminating Companies' results, previously mentioned.—Editor.] The work of Carpenter and Edwards would suggest the use of a copper-aluminum alloy if it could be easily and cheaply drawn into tubes, but the authors have never had an opportunity of experimenting with tubes of such material.

As regards the immediate future, and pending future investigations on different types of alloy, the best means at the disposal of engineers for hindering corrosion appear to be: (1) The entire abandonment of the use of plain 70:30 brass and its replacement by either the Admiralty alloy, 70:29:1 (tin), or Muntz's special brass, 70:28:2 (lead); (2) careful control of the temperature in condensers; (3) the extended use of electrochemical protection. The authors consider that the importance of temperature in determining the nature of the corrosive attack on brass by sea water has not received from engineers the attention it deserves.

The temperature of the water in a condenser can be kept down in the following ways: (1) By limiting the amount of steam condensed; (2) by shortening the path of the water through the condenser; (3) by increasing the speed of the water; (4) by preventing choking of the tubes, either by fitting screens or filters, or by frequent cleaning. The amount of steam condensed has been considered hitherto from the point of view of engine efficiency and space economy, and not at all from the corrosion standpoint. Nevertheless, it is a fact that, within the practical limits of temperature, the greater the amount of steam condensed the greater will be the tendency to selective corrosion, and if the condenser be really overloaded this form of corrosion may go on at an extremely rapid rate.

The practice of dividing a condenser into two "nests" is a distinct aid to corrosion. As the water passes through the lower nest it becomes heated, and consequently the top nest is supplied with water that is already warm and will become still more highly heated. From the corrosion point of view it would be preferable to introduce a separate water inlet and outlet to each nest of the condenser, and to discontinue the practice of building them in nests. The speeds of water through the condensers in the mercantile marine vary between 240 feet and 360 feet per minute. The average is about 290 feet per minute. The speed of the water used in the experimental plant was about 275 feet per minute. The faster the speed of the water in the condenser the lower the temperature, other things being equal, and the less the liability to the choking of tubes. In the authors' opinion the speed should be kept up at least to the highest figure mentioned above, and it might be worth while to try still higher speeds if the tubes frequently choke.

Inquiries made by the authors have shown that choked tubes are sometimes severely corroded, and sometimes not at all. If a choked tube be found in the bottom part of the condenser working normally it will not

usually be corroded, since its temperature, even in this case, is still low; if, however, the condenser is being overloaded, as shown by the high temperature of the outlet water, it may be severely corroded. If a choked tube be found in the top part, however, it will usually show severe dezincification, owing to the high temperature to which it has been heated. Hitherto the corrosion of choked tubes has been attributed to the electrochemical action of particles, more particularly of carbon, but this view has now been clearly shown to be untenable.

(NOTE.—In the above abstract many of the statements appear merely categorical, which are in reality not so. Space considerations have dictated the omission of many of the committees' arguments. Those who wish to read the report in full are referred to the transactions of the Institute of Metals, or to "The Engineer," Sept. 5, 1913, p. 263; Sept. 12, 1913, p. 287; Oct. 4, 1913, and "Engineering," Aug. 30, 1913, p. 302, and Sept. 5, 1913, p. 339).—"Power."

# SHIPS.

## UNITED STATES.

### UNITED STATES NAVAL VESSELS UNDER CONSTRUCTION.

PERCENTAGE OF COMPLETION.

| No | Vessel. | Building yard. | Engines. | No. shafts. | Speed, knots. | Percentage machinery completed 1914. | | Per ct. Hull com. Feb. 1, 1914. |
|---|---|---|---|---|---|---|---|---|
| | | | | | | Jan. 1 | Feb. 1. | |
| | BATTLESHIPS: | | | | | | | |
| 34 | *New York* | Navy Yard, N. Y. | Reciprocating | 2 | 21 | 94.95 | 96.10 | 96.0 |
| 35 | *Texas* | Newport News Co. | Reciprocating | 2 | 21 | 97.00 | 98.00 | 98 1 |
| 36 | *Nevada* | Fore River S. Co. | Curtis turbine | 2 | 20.5 | 60.31 | 62.46 | 62.6 |
| 37 | *Oklahoma* | New York S. Co. | Reciprocating | 2 | 20.5 | 60.58 | 63.08 | 56.3 |
| 38 | *Pennsylvania* | Newport News Co. | Curtis turbine | 4 | 21 | 1.90 | 2.42 | 14.4 |
| 39 | | Navy Yard, N. Y | Parsons turbine | 4 | 21 | ... | ... | 4 7 |
| | DESTROYERS: | | | | | | | |
| 45 | *Downes* | New York S. Co. | Curtis trb. & rec. | 2 | 29 | 89.75 | 92.82 | 93.7 |
| 50 | *Balch* | Wm. Cramp & Sons | Zoelly trb. & rec. | 2 | 29.5 | 96.11 | 96.71 | 92.6 |
| 51 | *O'Brien* | Wm. Cramp & Sons | Cramp trb. & rec. | 2 | 29 | 43.77 | 50.45 | 42.1 |
| 52 | *Nicholson* | Wm. Cramp & Sons | Cramp trb. & rec. | 2 | 29 | 43.18 | 49.41 | 42.1 |
| 53 | *Winslow* | Wm. Cramp & Sons | Cramp trb. & rec. | 2 | 29 | 42.68 | 50.05 | 40.3 |
| 54 | *McDougal* | Bath Iron Works | Parsons trb. & rec. | 2 | 30.5 | 70.60 | 75.70 | 66.6 |
| 55 | *Cushing* | Fore River S. Co. | Curtis turbine | 2 | 29 | 36.18 | 44.37 | 32.0 |
| 56 | *Ericsson* | New York S. Co. | Parsons trb. & rec. | 2 | 29 | 31.08 | 37.82 | 40.9 |
| 57 | | Fore River S. Co. | Curtis turbine | 2 | 29.5 | ... | 4.92 | 6.8 |
| 58 | | Wm. Cramp & Sons | Parsons turbine | 2 | 29.5 | ... | ... | 2 6 |
| 59 | | Wm. Cramp & Sons | Parsons turbine | 2 | 29.5 | ... | ... | 2.5 |
| 60 | | Bath Iron Works | Parsons trb. gear. | 2 | 30 | ... | 8.76 | 7.2 |
| 61 | | New York S. Co. | Parsons turbine | 2 | 29.5 | ... | 0.40 | 8.9 |
| 62 | | New York S. Co. | Parsons turbine | 2 | 29.5 | ... | 0.40 | 8.9 |
| | FUEL SHIPS: | | | | | | | |
| 13 | *Kanawha* | Navy Yard, Mare Isl'd. | Reciprocating | 2 | 14 | 15.60 | 17 90 | 31 5 |
| *14 | *Maumee* | Navy Yard, Mare Isl'd. | Diesel | 2 | 14 | ... | ... | 17 5 |
| | SUBMARINES: | | | | | | | |
| 31 | *G-3* | Lake T. B. Co. | Diesel-Sulzer | 2 | 14 | 30.00 | 30.00 | 80.3 |
| 26 | *G-4* | Wm. Cramp & Sons | Laurenti | 2 | 14 | 95.40 | 95.40 | 96.4 |
| 34 | *K-3* | Union Iron Works | Diesel | 2 | 14 | 97.15 | 99.86 | 91.7 |
| 35 | *K-4* | Seattle Con. & D. D. Co. | Diesel | 2 | 14 | 94.37 | 96.03 | 89.4 |
| 36 | *K-5* | Fore River S. Co. | Diesel | 2 | 14 | 92.44 | 95.82 | 88.9 |
| 37 | *K-6* | Fore River S. Co | Diesel | 2 | 14 | 91.18 | 95.38 | 88.7 |
| 38 | *K-7* | Union Iron Works | Diesel | 2 | 14 | 79 09 | 82.17 | 83.6 |
| 39 | *K-8* | Union Iron Works | Diesel | 2 | 14 | 77.56 | 81.89 | 83.0 |
| 40 | *L-1* | Fore River S. Co. | Diesel | 2 | 14 | 19.77 | 27.70 | 22.8 |
| 41 | *L-2* | Fore River S. Co. | Diesel | 2 | 14 | 19.06 | 27.70 | 22.6 |
| 42 | *L-3* | Fore River S. Co. | Diesel | 2 | 14 | 18.75 | 27.24 | 22.5 |
| 43 | *L-4* | Fore River S. Co. | Diesel | 2 | 14 | 18.25 | 27.24 | 22.4 |
| 44 | *L-5* | Lake T. B. Co. | Diesel | 2 | 14 | ... | ... | 7.4 |
| 45 | *L-6* | Lake T. B. Co. | Diesel | 2 | 14 | ... | ... | 0.0 |
| 46 | *L-7* | Lake T. B. Co. | Diesel | 2 | 14 | ... | ... | 0.0 |
| 47 | *M-1* | Fore River S. Co | Diesel | 2 | 14 | 18.80 | 24.44 | 14.0 |
| | SUBMARINE TENDERS: | | | | | | | |
| 1 | *Fulton* | Fore River S. Co | Diesel | 1 | 12.25 | 59.30 | 62 45 | 39.1 |
| 2 | *Bushnell* | Seattle Con. & D. D. Co. | Parsons turbines with reduc. gear. | 1 | 14 | 1.41 | 1.75 | 20.9 |
| | GUNBOAT: | | | | | | | |
| 19 | *Sacramento* | Wm. Cramp & Sons. | Reciprocating | 1 | 12.5 | 71.53 | 77 11 | 78.6 |
| | DESTROYER TENDER: | | | | | | | |
| 2 | *Melville* | New York S. Co. | Parsons turbines. | 1 | 15 | 1 50 | 2.36 | 29.6 |

* Engines being built at Navy Yard, New York.
Transport to be built at Navy Yard, Philadelphia. Supply ship to be built at Navy Yard, Boston.

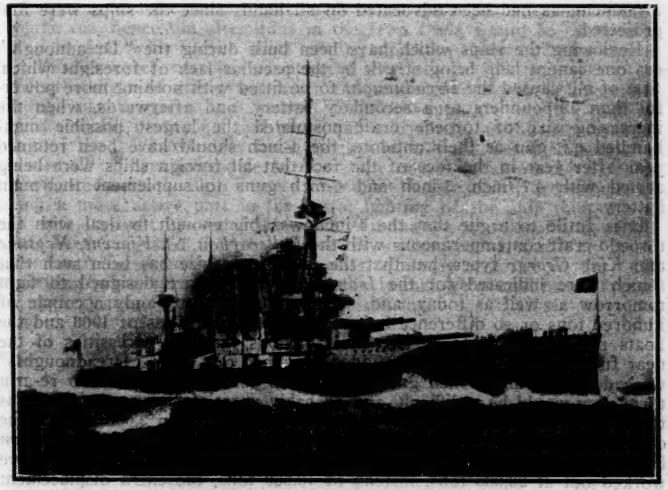
## CHILE.

### THE CHILIAN DREADNOUGHT ALMIRANTE LATORRE.

On November 27, 1913, the *Almirante Latorre*, the first of the two Dreadnought battleships which Sir W. G. Armstrong, Whitworth & Co., Ltd., are building for the Chilian Government, was successfully launched at the Elswick shipyard. The *Almirante Latorre* has the following approximate dimensions:

| | |
|---|---|
| Length, overall, feet | 661 |
| between perpendiculars, feet | 625 |
| Breadth, feet | 92 |
| Depth, molded, feet and inches | 43.9¼ |
| Mean draught, feet and inches | 28.6 |
| Displacement, tons | 28,000 |

The *Almirante Latorre* and *Cochrane* will not only be the largest and most powerful battleships in the Chilian Navy, but they will rank as the largest vessels yet constructed by Sir W. G. Armstrong, Whitworth & Co., Ltd. The main propelling machinery of the *Almirante Latorre* is being supplied by Messrs. John Brown & Co., Ltd., of Clydebank, and consists of two sets of Parsons marine turbines working on four shafts, and of approximately 37,000 shaft H.P., sufficient to give the battleship a speed of 23 knots. There are twenty-one boilers of the Yarrow tube type (pressure 250 pounds per square inch). Both coal and oil fuel will be used. The main armament comprises ten 14-inch guns, whilst the secondary armament will consist of sixteen 6-inch guns, four 3-inch guns, two 76-mm. 12-pdr. boat guns, and four Maxim machine guns, or a total main and secondary armament of thirty-six guns and four 21-inch submerged torpedo tubes. The armament is being manufactured by Sir W. G. Armstrong,



CHILIAN BATTLESHIP "ALMIRANTE LATORRE."

Whitworth & Co., Ltd., at whose works at Openshaw are also being manufactured the armor plates employed in this battleship. The complement of the ship will be 1,075 men. The *Almirante Latorre*, named after a distinguished officer of the Chilian Navy who died some two years ago, is the ninth war vessel constructed by the Armstrong firm for the Chilian Government. The first was the *Esmeralda*, of about 3,000 tons, built at the Walker shipyard in 1884. The others were the *Blanco Encalada, Ministro Zenteno*, the second *Esmeralda, O'Higgins, Chacabuco, Captain Thomson* and *Jenera Baquedano*, all of which were built at the Elswick shipyard. Inclusive of the *Almirante Cochrane*, their displacement aggregates 90,000 tons, with about 159,000 horsepower.—"Shipbuilding and Shipping Record."

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## ENGLAND.

### H. M. DREADNOUGHT *IRON DUKE*.

The four battleships of the "Iron Duke" class, of which the name ship, built at Portsmouth, is now completing for sea and the last, the *Emperor of India*, took the water at Barrow at the end of November, are not so much developments of the "King George V" class as examples of what those ships should have been but for the conservative policy of retaining the 4-inch guns in them instead of fitting 6-inch as an anti-torpedo craft armament, as had been advocated on all hands since the ships were first projected.

Reviewing the ships which have been built during the "Dreadnought" era one cannot help being struck by the peculiar lack of foresight which, first of all, caused the *Dreadnought* to be fitted with nothing more powerful than 12-pounders as a secondary battery, and afterwards, when the increasing size of torpedo craft postulated the largest possible manhandled q.f. gun as their antidote, the 4-inch should have been retained year after year in the face of the fact that all foreign ships were being armed with 4.7 inch, 5-inch and 6-inch guns to supplement the main battery.

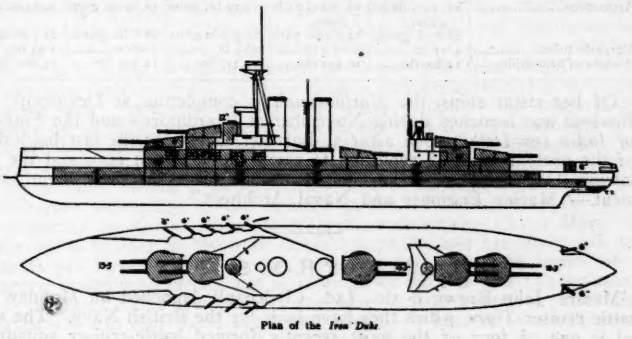
It is futile to argue that the 4-inch was big enough to deal with the torpedo craft contemporaneous with the *Bellerophon, St. Vincent, Neptune* and *King George* types, but that the increase in size has been such that 6-inch were indicated for the *Iron Duke*. Ships are designed to fight tomorrow as well as today, and, seeing that there are only a couple of hundred tons or so difference between the big tribal class of 1906 and the boats now passing into service, it is obvious that the necessities of the near future could have been provided for when our early "Dreadnoughts" were designed, and 6-inch guns included in their equipment. To re-arm them now would be a difficult, and probably unsatisfactory, proceeding, as the substitution of 6-inch for 4-inch would add close on 2,000 tons to the displacement—in theory. That is to say, supposing that the *Bellerophon* had been fitted with 6-inch in the original design her tonnage would have worked out at 20,600 tons, instead of 18,600 tons, the extra displacement being absorbed in additional weight of guns, fittings, ammunition, armor and engine power to maintain speed. If, therefore, these pre-"Iron Dukes" were given 6-inch now the alterations would probably affect the speed and internal economy of the ships so much that the game would not be worth the candle.

In general appearance the *Iron Duke* greatly resembles the ships that preceded her. The disposition of the five turrets are the same as in the "King George V" class, and the profile is only different through modifications in the rig and funnels. Instead of the pole mast forward, with

light tripod struts, the *Iron Duke* has the heavy tripod mast of the earlier all-big-gun ships, while the two funnels are both round and of the same size, instead of the foremost being oblong and double the diameter of the after one. Considering that the horsepower of these later ships is in excess of any previous battleships, it is evident that they are intended to realize their power more by the use of oil than coal fuel, otherwise the uptakes would have been much bigger than they are.

The armament consists of ten 13.5-inch guns, firing a 1,400-pound projectile, and twelve 6-inch. The former are mounted two forward, the second turret being superposed, one amidships and two aft, with the fourth firing over the fifth. All ten can thus be trained on either beam, and four in the keel line ahead or astern, the total broadside being 14,000 pounds as against the 6,800 pounds of the *Dreadnought*. The turrets have 12-inch faces and 8-inch backs, the bases being 10 inches. Of the smaller guns ten are disposed up forward along the upper deck, the forecastle being recessed so as to allow of their being trained well ahead, as shown in the plan. Right aft a couple are carried low down on the main deck and in recessed ports to secure a dead-astern bearing. The exact reason why they should have been so placed is uncertain, as the command is low and likely to be interfered with in rough weather. If desired they could have been accommodated in the after superstructure, as are the 4-inch in the *King George V*, where a better all-round bearing would have been allowed and any weather would not have interfered with their handling. The explanation seems to lie in the fact that at night torpedo craft can be best spotted from positions low down on the water line, when they show up against the sky, and not from an elevation, when they merge into the sea, hence the after guns in the *Iron Duke* would be excellently placed to deal with hostile boats coming up from astern. Other causes than this, however, may have influenced the design.

The thickness of the armor over the secondary battery is uncertain, although usually given as 8 inches. If 6-inch guns are worth mounting they are worth protecting, and protecting adequately, as the experimental firing of the "Town" class with 6-inch and 4.7-inch pieces showed that these small calibers were able to wreck the upper works of the *Empress of India* and turn the ship into a furnace, and consequently are capable of taking a more active part in the battle fighting of the ship than the 4-inch with its light 31-pound projectile. In the absence of official details it has been suggested that this battery is unarmored after the U. S. A. system introduced in the "Nevada" class, which is armored on the "all-or-nothing" plan, but it is more than unlikely that we should have adopted any such system for the ships under discussion.



Plan of the *Iron Duke*

PLAN OF THE "IRON DUKE."

The dimensions are: Length, 575 feet between perpendiculars and 620 feet over all; beam 89½ feet and mean draught 27½ feet, the designed displacement being 25,000 tons. The *Iron Duke* is thus 24 feet longer over all than the *King George V*, and exceeds her tonnage by 2,000 tons—the cost of including the 6-inch guns. Driven by Parsons turbines, having a designed H.P. of 33,000, the speed is to be 21.5 knots, but this is likely to be exceeded by more than one knot.

As may be seen from the plan, the hull is very adequately protected, there being almost complete belts along the water line and lower deck side, and a thick strake running from the fore to the aftermost turret along the main deck. Amidships the water-line belt is 12 inches, tapering to 6 inches towards the extremities and falling short of the cut-water and stern by about 20 feet. Above this, the lower-deck side is similarly armored, except that the belt ends a little further from the stern; along the main deck the thickness is 10 inches, with a probable 8 inches over the batteries. As is customary in our ships there is a 2¾-inch protective deck, which reinforces the main belt and encloses the machinery and vitals.

Forward there is a large 12-inch conning tower with a small observation station aft. Abaft the conning tower forward is a three-decked pile of chart house, cabins and bridges, which is quite as big as that in the *Lion*, and from which the fore funnel is well spaced. In front of the second funnel is a stump mast, from which the boat derrick is swung, and to the head of this the w.t. yards are fitted to sustain the aerials from the foretopmast.

Four torpedo tubes are fitted, and these will discharge the 21-inch weapon. Submarine telephones are also installed, and a special feature is the protection against air craft.

The *Iron Duke* compares most favorably with her contemporaries abroad, as may be seen from the following table:

| | England. *Iron Duke.* | Germany. *Konig Albert* | France. *Courbet.* | Austria *P. Eugen.* | U. S. A. *Texas.* |
|---|---|---|---|---|---|
| Length | 575 feet | 564 feet | 541½ feet | 495½ feet | 565 feet. |
| Beam | 89½ feet | 95½ feet | 88½ feet | 89½ feet | 95½ feet. |
| Draught | 27½ feet | 29½ feet | 27½ feet | 26½ feet | 28½ feet. |
| Displacement | 25,000 tons | 24,310 tons | 23,026 tons | 20,000 tons | 27,000 tons. |
| Designed H.P | 33,000 | 28,000 | 28,000 | 25,000 | 28,000. |
| speed | 21.5 knots | 21 knots | 20 knots | 20.5 knots | 21 knots. |
| Armor belt | 12. in. 6 in. | 12 in. | 10½ in | 11 in | 9 in. 12 in. |
| Above | 12 in. 10 in. | 6½ in | 7 in | 7½ in | 8 in. 10 in. |
| Deck | 2½ in | 3 in. | 2½ in | 3 in. | 3 in. |
| Big guns | 13 in. | 10 in. 13½ in. | 10½ in | 8 in. 11 in. | 8 in. 11 in. |
| A.T.C. | 8 in | 6½ in. | 7-in | 6½ in | 8 in. |
| Armament | 10 13.5-inch guns. 12 6-in. guns. | 10 12-inch guns. 14 5.9-in guns | 12 12-in. guns 22 5.5-in.guns | 12 12-in. guns 12 6 in. guns. | 10 14-in. guns. 21 5-in. guns. |
| Torpedo tubes | 4 21-in | 5 19.9 in | 4 18 in | 4 18-in | 4 21-in. |
| Weight of broadside | 14,600 lbs | 10,442 lbs | 10,427 lbs | 12.314 lbs | 14,660 lbs. |

Of her sister ships, the *Marlborough* is completing at Devonport, the *Benbow* was launched during November at Beardmore's and the *Emperor of India* (ex *Delhi*) soon after at Barrow. They are the last battleships of the present cycle of evolution to carry the 13.5-inch gun, and the following "Warspite" class have eight 15-inch pieces as their main armament.—"Marine Engineer and Naval Architect."

## LAUNCH OF H. M. S. *TIGER*.

Messrs. John Brown & Co., Ltd., Clydebank, launched on Monday the battle cruiser *Tiger*, which they have built for the British Navy. The vessel is one of four of the most recently formed battle-cruiser squadron.

The three other ships launched some time ago and now in commission are the *Lion*, the *Princess Royal* and the *Queen Mary*. All the vessels will be capable of maintaining for very long periods a speed of 28 knots under easy conditions alike in the stokeholds and the turbine rooms. In ordinary circumstances coal will be used, but the boilers are adapted also for burning oil fuel, and as provision is made for carrying a large quantity of liquid fuel, it follows that the maximum speed will be maintainable for a much longer time than would otherwise be the case.

The four ships are of the same length, 660 feet, but in order that the draught might remain the same, notwithstanding the increased displacement due to greater fighting power, the beam in the case of the *Tiger* has been augmented. To insure the same speed with greater displacement tonnage, the power of the boilers and turbines has been increased; but there is no intention, it is understood, of attaining with the *Tiger* a higher speed under normal conditions than the 28½ knots realized by the preceding ships of the class.

The emplacement of the guns is the same in all four ships, so that to all intents and purposes their tactical qualities are alike. The *Tiger* is fitted with 12 6-inch guns, all of them with armor protection and arranged specially for end-on fire; whereas in the earlier cruisers of the class the torpedo-repelling armament included 16 4-inch guns.

In the *Tiger* the Brown-Curtis turbine has been adopted, while in the other vessels the turbines are of the Parsons type. The turbines on each of the four shafts constitute a separate unit, each shaft having high-pressure impulse and low-pressure reaction turbines, along with astern turbines.

It is believed that the *Tiger's* turbines will be the most powerful ever constructed. It is stated that they will develop over 100,000 shaft H.P., and that the vessel will be the fastest armored warship afloat. The launch took place shortly after noon, the naming ceremony being performed by Lady Helen Vincent.

The launching weight of the hull was 13,000 tons and the time required for passing down the slips was 60 seconds. The vessel was berthed in the fitting-out basin under the large hammer-head crane, at the opposite side of the basin from where the *Aquitania* is berthed.

## ALL-OIL BATTLESHIP.

Britain can take credit in the fact that it possesses the first "all-oil" battleship in the world, for the *Queen Elizabeth* will carry no coal whatsoever. We have, says "The Petroleum Review," a large number of destroyers so equipped, but the *Queen Elizabeth* is the first "all-oil" battleship to be launched, although the American *Nevada* and *Oklahoma* were the first to be designed. The substitution of oil for coal not only gives the ship a radius of action greater by 60 per cent. than would otherwise have been the case, but it will also allow a very great reduction to be made in the crew. In a coal-burning ship nearly a half of the complement is made up of stokers, who have to bring the coal from the bunkers to the stokehold, and shovel it in as it is required. In the oil-burning ship, however, the fuel is pumped from the tanks and double-bottoms straight into the furnaces, where it burns fiercely at the end of a spray. The exact difference oil makes to the fighting power of a ship may be seen by comparing the *Queen Elizabeth* with the coal-burning *Queen Mary*. The former is heavier by 500 tons, displacing 27,500, and her speed will be 28 knots per hour. The battleship, however, will have 13½ inches of side armor as against 9 inches, and a broadside of eight 15-inch (15,600 pounds) as against eight new-pattern 13.5-inch (11,000 pounds). The *Queen Elizabeth* will cost about £2,250,000—roughly, a quarter of a million more than the *Queen Mary*.

## FRANCE.

### THE MINE TRANSPORTERS *CERBERE* AND *PLUTON*.

NOTE.—These vessels represent a type that has not yet been built for our Navy. Their general characteristics are of interest. They apparently represent a type of vessel for which there is great need if mines are to be extensively used. The following is taken from "Le Yacht":

These vessels are the first two to be specially constructed for the employment of submarine mines. Until now we have been content to modify some of our old destroyers, which are at best not well fitted for this service and which can carry but a very inadequate supply of these terrible engines of destruction.

Their characteristics are as follows:

| | |
|---|---|
| Displacement, loaded, tons | 594 |
| for trial, tons | 541 |
| Length, total, feet | 199.4 |
| between perpendiculars, feet | 193.5 |
| Beam, feet | 27.23 |
| Depth, feet | 14.17 |
| Draught, feet | 10.66 |

The hull is constructed of steel. The metal keel is protected by a teak keel. There are five longitudinals in the central portion of the vessel and three at the ends. There are two rolling keels. The hull is divided transversely into six watertight compartments. Contrary to the usual custom, the crew are berthed aft, while the officers' quarters are forward.

The following table gives the particulars of the machinery and the results of the trials:

| | *Cerbère.* | *Pluton.* |
|---|---|---|
| Built by | Chantiers de Bretagne | Ch. A. Normand. |
| Boilers | Two du Temple | Two Normand. |
| Grate surface, feet | 149.9 | 148.5. |
| Heating surface, feet | 575.4 | 693.7. |
| Pressure, pounds | 256 | 256. |
| Blowers | Two Rateau, vertical | 2 Normand alt. motors, |
| Bunker capacity, tons | 145–190 | 145–190. |
| Engines | Two triple-expansion | Two triple-expansion. |
| Cylinders, inches | 18.9, 29.13, 45.6 | 18.9, 29.13, 45.6. |
| I.H.P | 3,000 | 3,000. |
| Propellers, diameter, ft. | 8.2 | 8.2. |
| Speed, knots | 20 | 20.87. |

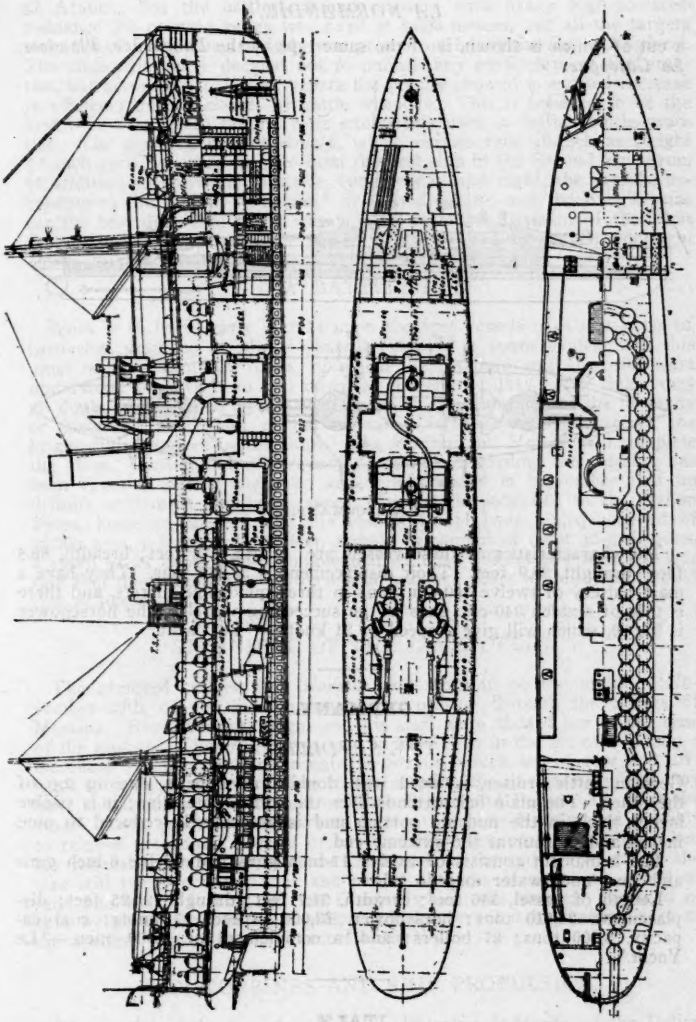
Electricity is furnished by a 4-kw. dynamo at 80 volts.

The bunkers give some protection, being arranged outboard of boiler-room and engine-room spaces. When fully loaded the vessel will carry 190 tons of fuel, which will give a theoretical radius of 300 miles at full power and 1,900 miles at 16 knots.

This type of vessel have need of the very best maneuvering qualities. They have two rudders, one in the bow and one in the stern, have surfaces of 49.5 and 23.6 square feet.

The armament is composed of a 75-mm. rapid-fire gun forward and 140 submarine mines. These mines are placed on four tracks under the main deck. They are put overboard by two large davits located on each side aft.

Each of these vessels will carry ten times the load of one of the transformed destroyers.



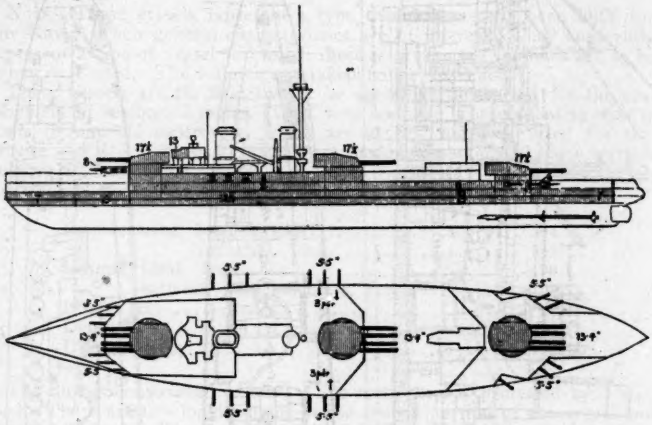
FRENCH MINE PLANTER.

These mines are arranged to float at a depth of four meters below the surface, and the draught of the vessels is regulated so as to allow them to pass over the mine field without danger.

The results obtained in actual service have been so satisfactory that additional units of this type have been put in construction.

*LA NORMANDIE,*

a cut of which is shown, is of the same type as the *Lauguedoe, Flanders, La Gascogne.*



"LA NORMANDIE."

The characteristics of these vessels are: length, 573 feet; breadth, 88.8 feet; draught, 28.9 feet. Their displacement is 25,250 tons. They have a main battery of twelve 340-cm. guns in three quadruple turrets, and there is talk of sixteen 340-cm. guns in the succeeding vessels. The horsepower is 33,000, which will give a speed of 21 knots.—"Le Yacht."

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## GERMANY.

THE *SEYDLITZ,*

German battle cruiser, is fitted with double armor belts, one on top of the other. The main belt extends over the entire water line; it is twelve inches thick in the midship portion and is successively reduced to nine inches and to four at the extreme end.

The armament consists of twelve 11-inch guns and twelve 6-inch guns and four underwater torpedo tubes.

Length of vessel, 346 feet; breadth, 91.7 feet; draught, 26.25 feet; displacement, 24,610 tons; horsepower, 63,000; speed, 27 knots; coal capacity, 3,600 tons; 27 boilers; and a complement of 1,000 men.—"Le Yacht."

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## ITALY.

BATTLE PRACTICE.

It has been stated that the autumn battle practice of the Italian Fleet was carried out this year on a larger scale than ever before. Every commissioned ship took part in the firing, the conditions for which were more severe than on former occasions. The scene of the practice was the Gulf

of Aranci. For the battleships and cruisers with heavy high-powered ordnance the opening range was fixed at 8,000 meters, and all the targets were towed. The King of Italy was present on board the *Dante Alighieri.* The authorities have decided not to publish any particulars of the practice, but according to Italian papers the results showed a marked increase in efficiency for all classes of battle weapons. This is believed to be the first time that towed targets were exclusively used in Italian battle practice. The armored cruiser *Amalfi,* which carries four 10-inch and eight 7.5-inch guns, is stated to have been the best ship in the Second Squadron. In addition to individual practice, both by day and night, the vessels, according to the Roman "Tribuna," fired in divisions, and the King's prize for the best divisional firing was won by the First Division of the First Squadron, which includes the newest ships, headed by the Dreadnought *Dante Alighieri.*

## NEW BATTLESHIPS.

Work is to begin early in 1914 upon the first vessels of a new class of battleship, designed by Major-General Ferrati. It seems evident that this must be the "Dandolo" class, of which the *Dandolo* and *Morosini* were understood to have been laid down at the end of 1912. The delay may no doubt be explained by a lack of financial provision, and to the necessity of installing new plant in the ordnance factories to manufacture the larger guns required by the ships. The *Mazzini* and *Mameli* will complete the class. Considerable divergency of opinion regarding their design has been revealed during the year, and it was stated in November that no definite settlement had been made. The details published in the Italian Press, however, showed that the vessels would have a displacement of 30,000 tons, a length of 692 feet, a main armament of eight 15-inch guns, a speed of 24 knots, and a complement of 1,300 officers and men. It was said that they would take between four and five years to build, and would cost £3,400,000 each.

## STRANDING OF THE *SAN GIORGIO.*

The armored cruiser *San Giorgio* went aground on the night of November 20th, on the Sicilian coast after passing through the Straits of Messina. Rear Admiral Bagni and his staff were aboard her at the time of the mishap. The cruiser was said to have been in the act of avoiding a merchant steamer passing through the Straits, which are narrow and difficult to navigate, and according to one account, the navigating officer may have confused the light houses of Cape Peloro and Pezzo. To salve the vessel it was reported necessary to lighten her considerably, and possibly to remove the guns. Several of her watertight compartments filled when she ran ashore at the speed of 14 knots, and at the end of November she was still lying in about four and a half fathoms of water on a bed of sand and rock. It was the *San Giorgio* which was severely damaged in 1911 by going aground at Posillipo, which prevented her taking part in the opening phases of the Turco-Italian War.

## TOSI TURBINES AND SHIP PROPULSION.

The results of the tests to which the destroyer *Indomito,* of the Italian fleet, fitted with Tosi turbines, had been subjected, have already been published. Since then other torpedo-boat destroyers have been subjected to similar tests.

The following are the results obtained with three Italian torpedo-boat destroyers recently completed:

| | *Indomito.* | *Intrepido.* | *Irrequieto.* |
|---|---|---|---|
| Average speed at full power, knots..... | 35.068 | 35.040 | 35.796 |
| Power developed in E.H.P..... | 17.080 | 17.730 | 17.640 |
| Maximum speed, knots..... | 35.630 | 36.740 | 36.039 |
| Number of revolutions, at average speed with full power..... | 690 | 692 | 696 |
| Consumption of naphthaline per H.P. hour at full speed..... | Average 0.590 kg. | | |

The time for reversing the engine for a course in the opposite direction at varying speeds of 400, 500 and 600 revolutions at full power was also recorded, and the following are the results:

For passing from "ahead" to "astern," the time was from 9 to 14 seconds, according to the speed of the ship, and for passing from "astern" to "ahead," the time was from 5 to 7 seconds, again according to the speed of the ship.

It was noticed that when passing from full power ahead to full power astern, the latter movement reached a maximum of 400 r.p.m.

On account of the results obtained with the Tosi turbines, the Italian Government, in placing an order with the Odero and Pattison shipbuilding companies for eight new torpedo destroyers of 700 tons, of the "Francesco Nullo" type, specified that these turbines should be utilized as the means of propulsion.

Quite recently the Pattison yard, Naples, gave an order to the latter firm, for eight other turbines, each of 20,000 E.H.P. ahead movement and 6,500 E.H.P. astern movement, with 500 revolutions, intended for four destroyers for the Roumanian fleet, of 1,500 tons, and a speed of 35 knots.

The weight of each turbine per E.H.P. is from 3.25 kg. to 3.50 kg.

The following are the principal measurements of the Italian torpedo-boat destroyers:

| | |
|---|---|
| Length, feet ..... | 239½ |
| Breadth, feet ..... | 23 |
| Average draught, feet..... | 7½ |
| Displacement during trial, tons..... | 612 |
| fully equipped and manned, tons.... | 620 |
| Maximum speed, knots..... | 35 |

The principal measurements of the Roumanian destroyers, however, are as follows:

| | |
|---|---|
| Length, feet ..... | 308½ |
| Breadth, feet ..... | 30½ |
| Average draught, feet..... | 9¼ |
| Displacement during trial, tons..... | 1,330 |
| with full equipment and manned, tons. | 1,450 |
| Maximum speed, knots..... | 35 |

The accompanying photographs represent a destroyer at full speed, and a Tosi turbine of 9,000 E.H.P. installed on the illustrated destroyer.

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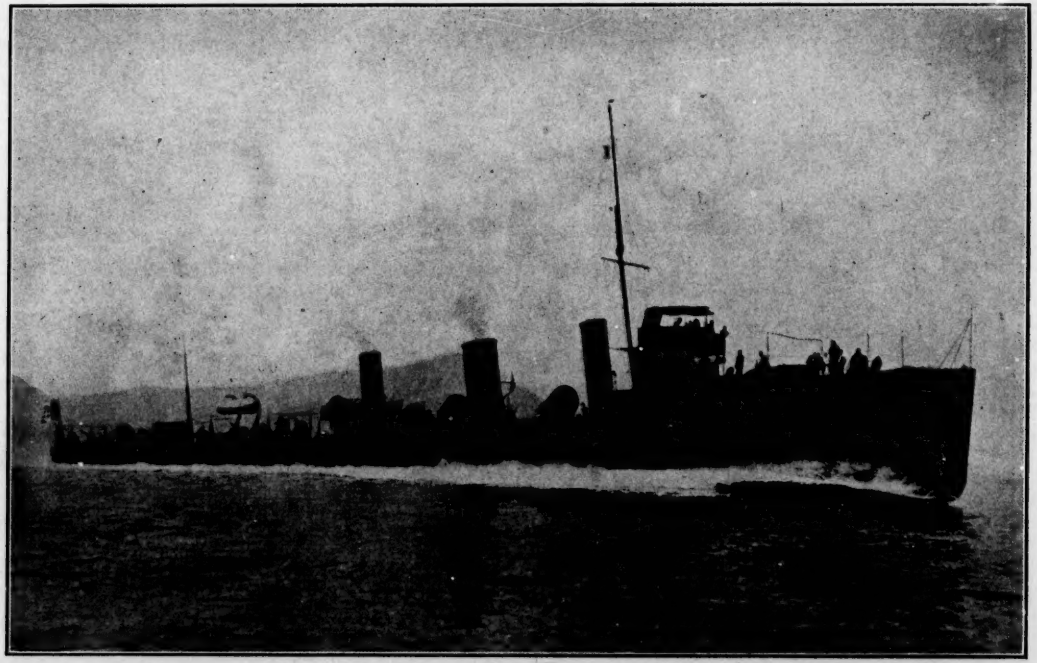
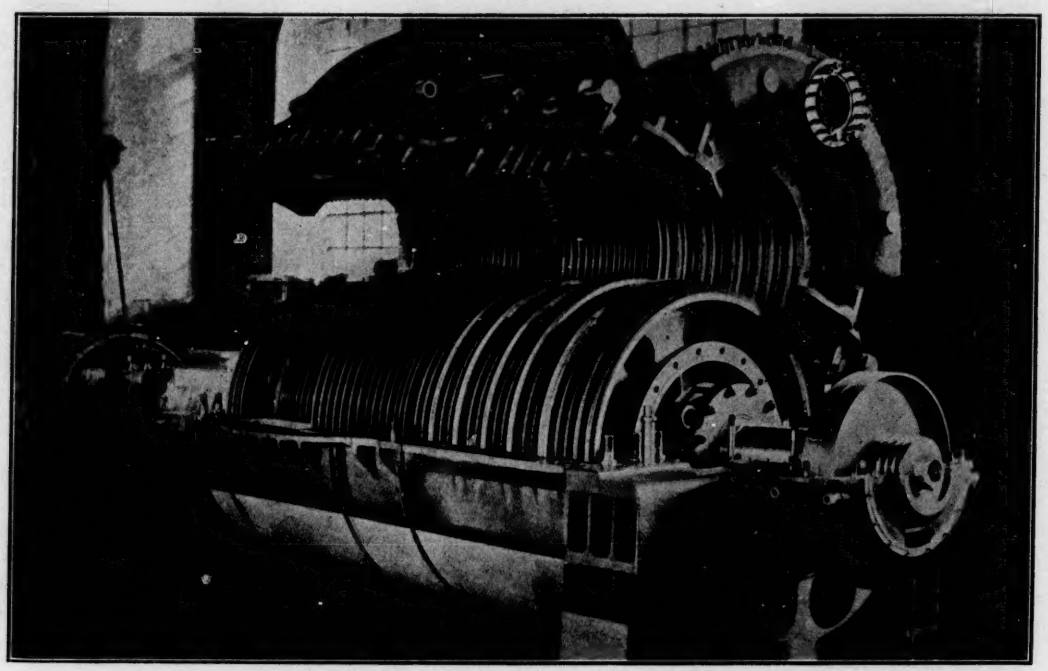
## JAPAN.

### NEW BATTLESHIP BEGUN.

A Reuter telegram from Tokio on November 20th, announced the beginning at Yokosuka of the fourth battleship of the "Fuso" class, and it was stated that the new vessel will have a displacement of 30,600 tons. With the laying down of this ship Japan has twelve Dreadnoughts built



ITALIAN DESTROYER "INDOMITO" ON FULL-SPEED TRIALS.



TURBINE 9,000 E.H.P. OF ITALIAN DESTROYER "INDOMITO."

and building. Four battleships of the "Satsuma" and "Kawachi" types are completed, as is the battle cruiser *Kongo*, which arrived in Japanese waters on November 5th. Three sister ships to the *Kongo* are in hand. Lastly, there is the class of four "super-Dreadnoughts," to which the new ship at Yokosuka belongs. These latter vessels have been designed for 23 knots speed. Their armament has been given by some accounts as twelve 14-inch, and by others as ten 15-inch guns. They will also carry sixteen 6-inch guns.

### SUBMARINE LAUNCHED.

A submarine of the "Lauboeuf" type was launched for Japan at Chalons-sur-Saone, France, on November 11th.

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## RUSSIA.

### FIRST BLACK-SEA DREADNOUGHT LAUNCHED.

The battleship *Imperatritsa Maria* was launched at Nikolaieff on November 1st. She was the first of the three Dreadnoughts laid down in 1911–12 for service in the Black Sea to take the water. The design of the trio differs in some details from that of the four Baltic Dreadnoughts launched in 1911 and now completing, although the size of the two classes is practically the same. Instead of 16 4.7-inch guns, the Black-Sea vessels have 20 5-inch in their anti-torpedo-defence armament; their speed is 21 instead of 23 knots; their bunker capacity is smaller; but there is an increase in armored protection. The main armament of twelve 12-inch guns is to be carried in four triple turrets in both classes.

### GUNNERY PRACTICE.

Experiments have recently been carried out by vessels of the Black Sea Fleet with the obsolete battleship *Tchesmé* as a target. The damage caused by the firing caused the ship to founder. A 12-inch shell which entered slightly below the waterline amidships cut a clean hole through the 16 inches of compound armor which the *Tchesmé* carried, and sent her to the bottom. The *Tchesmé* was launched at Sebastopol in 1886. The armored cruiser *Bayan* has received the prize offered by the Tsar for the ship holding the gunnery record in the Baltic Fleet.

## BOOKS RECEIVED.

SCREW PROPELLERS AND ESTIMATION OF POWER FOR THE PROPULSION OF SHIPS. By CAPTAIN CHARLES W. DYSON, U. S. Navy. Published by JOHN WILEY & SONS, New York. 137 pages of text, with an Atlas of 32 plates. Price $7.50 net.

This work is the culmination of about twelve years of patient investigation and careful study on the part of the author. The results of these investigations have appeared from time to time in the form of contributions to this Journal. The first of these appeared in the May, 1903, number, and contained in tabulated form analyses of the performance of the propellers of various naval vessels, including all of our own and several belonging to other nations. In analyzing their performances the formulas of Froude and S. W. Barnaby were applied, the latter having first been reduced to the form of a logarithmic chart. In addition the article included discussion showing the effect of variation in wake coefficient, and variations in propeller surface, diameter and pitch. The data given was of very wide scope, sufficient to enable a fairly good propeller to be picked out for a vessel by direct comparison.

In the February, 1909, number appeared a second paper, bringing the data of the previous paper up to date, including the earlier turbine vessels, and containing the results of investigations into the form of blade that is most efficient. Curves were given for various vessels, including indicated and effective horsepower, propulsive coefficients, indicated thrust, and the developed and projected areas.

In August, 1910, the third paper appeared, giving more particularly the results of Captain Dyson's investigations into

the relations between effective horsepower as deduced from the model-tank experiments and shaft horsepower, with the effect of appendage resistance. The chart method of design is introduced in this paper.

Further investigation induced to some extent by discussion of the last paper led to a continuation of the subject in November, 1910. In this paper the chart of design is extended to take into account the influence of tip speed, and the method of application is explained to include propellers of widely varying power and speed of revolution.

In a fifth paper, appearing in the August, 1911, number, the methods of Captain Dyson appear to be rounding into the finished shape in which they appear in the volume now under discussion. The tabulated, basic data is perfected and the charts of design are amplified with several problems to illustrate the method of application. The standard form of blade used by the U. S. Navy is explained, and methods are given for determining particulars of different propellers that will give equal efficiencies, with reasons for making choice between them.

Further study of the methods employed led Captain Dyson to greatly simplify the charts shown in the last-named paper, with a corresponding decrease in the work necessary in using them to arrive at correct results. The results of this study are contained in a last paper appearing in the May, 1913, number of the Journal, in which the whole field of propeller analysis is gone into, using the curves of E.H.P. derived from model experiments as a basis. This includes the proper correction for block coefficients, the effect of "thrust deduction" or "augment of resistance," terms applied to the peculiar effect upon thrust of the propeller exercised by fullness of the after-body lines of the hull, and the effect of the various appendages to the bare hull. The prevailing method of design, using theoretical formulas and by "direct" comparison, are briefly discussed and their errors are pointed out.

This review of the field of labor that led finally to the publication of Captain Dyson's book is necessary for a full

appreciation of the practical merits of the work. The accuracy of the methods laid down may be understood from the following concrete example. The propeller designed by Captain Dyson for the Battleship *Texas* was calculated to give a speed of 21 knots at 125 revolutions, with 28,100 indicated horsepower. The actual corresponding results on the four-hours full-speed trial were 21.05, 124.56 and 28,373.

The text of the book contains as an introductory a short history of the development of screw-propeller propulsion. Then follow the chapters giving account of the various factors entering into questions of design, various theories and methods of design by comparison are discussed and their defects pointed out. The standard forms of blades are next discussed with the effect of varying pitch in the case of adjustable blades and the effect on performance of the propeller by varying any of its elements. Materials of construction and the design of the propeller for strength are also discussed.

Chapter IX is taken up with an explanation of the design charts given in the Atlas. Various problems are discussed in Chapters X to XII and forms for computation are given. Chapter XIII discusses the subject of cavitation from the author's point of view, his conclusions being reached in the words "having two sections of different form and offering, approximately, the same resistance to motion through the water, that section which when applied to a screw propeller produces the greater increase of actual over nominal pitch will produce the earlier cavitation."

Chapter XIV, contributed by Mr. Luther D. Lovekin, explains the geometry and method of drafting the screw propeller. Chapter XV concludes the book with a list of the charts and other sheets embraced in the Atlas. These are as follows:

Sheets 1 to 7—Propeller projected-area forms of a large number of U. S. Naval vessels with additions to include vessels of every class.

Sheets 8 to 11—Actual projections, including particular data of 38 propellers used in the U. S. Navy.

Sheets 12 to 15.—Ship and propeller data tabulated, including constants worked out for use with the charts, for all classes of vessels.

Sheet 16—Computation form, including the author's formulas for making computations.

Sheets 17 to 25—Design charts as follows:

Chart I. Corrections for block coefficients, including corrections for shape of hull, appendages, etc.

Chart II. Appendage resistances.

Chart III. Curves of propulsive coefficients and indicated thrust for different values of ratio of projected area to disc area.

Chart IV. Curves of tip speed and indicated thrust.

Chart V. Chart of design for maximum allowable thrust curves.

Chart VI. Chart of design for minimum allowable thrust curves.

Chart VII. Curves of corrective factors for thrust deduction.

Chart VIII. Standard projected-area forms.

Chart IX. Combination chart, numbers 1, 7, 2, 5 and 6 laid down in circular form in order that revolving scale pivoted at center may be used.

Sheet 26. Geometrical construction of propeller blade, vertical generatrix.

Sheet 27. Geometrical construction of propeller blade, inclined generatrix.

Sheet 28. Drawing of propeller, merchant type.

Sheet 29. Drawing of propeller, destroyer type.

Sheet 30. Drawing of propeller, battleship type.

Sheet 31. Chart showing increase of projected area necessary for overloads at constant speed of vessel.

Sheet 32. Barnaby chart of propeller efficiencies.

While making his investigations Captain Dyson has been on duty in the Bureau of Steam Engineering, Navy Department, and his methods are now employed and relied upon by the Department.

His book offers no new theories on propeller design. The methods deduced combine the best theories of other investigators with the author's painstaking detailed investigations into actual propeller data extending over a period of more than twelve years. The methods of analysis that are laid down have been proven to give an accurate forecast of the performance of a propeller. The work will serve as a monument to the zealous labors and analytical powers of the author. The text is in clear type and printed in attractive style. The plates are handsomely engraved and are bound as whole sheets with a single fold. The work represents the latest thought and best practice in propeller design and analysis, and should be in the hands of every marine engineer.

An admirable statement of the correct principles which form the basis of propeller design will be found in an editorial in "The Engineer" (London) of December 12, 1913, page 629. This is recommended to those interested as a very clear exposition of the problems to be solved and the practical difficulties to be overcome, all of which are taken into account in Captain Dyson's book.—U. T. HOLMES.

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A READER OF SCIENTIFIC AND TECHNICAL SPANISH, FOR COLLEGES AND TECHNOLOGICAL SCHOOLS, WITH VOCABULARY AND NOTES. By CORNELIS DEWITT WILLCOX, Lieutenant Colonel United States Army, Professor U. S. Military Academy. Late General Staff Corps. Published by STURGIS AND WALTON, New York.

This book has been prepared for use of students in technical schools and colleges who desire to prepare themselves for the practice of the Engineering Profession in Spanish-speaking countries.

It contains short descriptions in Spanish, of various engineering apparatus, installation and systems, the reading of which will give not only a fair idea of Spanish technical language, but will serve to give a working knowledge of it.

The idea of thus acquiring familiarity with the use of Spanish technical terms is somewhat novel and should be quite effective in its purpose.

Five hundred and eighty-eight pages, sixty-seven illustrations.

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PRACTICAL USES OF THE WAVE METER IN WIRELESS TELEGRAPHY. Book by J. O. MANBORGNE, 1st Lieut. 24th U. S. Infantry. Published by McGRAW HILL BOOK Co., New York. Seventy-five pages. Price, $1.00 net.

This book offers descriptions of wave-measuring devices and methods of making radio measurements. It should be of great assistance to installers and operators and particularly to experimenters. The text is given in clear and simple language and well illustrated by simple illustrations and diagrams.

# ASSOCIATION NOTES.

## MEETING DECEMBER 30, 1913.

A meeting of the Society was called on December 13, 1913, at which time the ballots cast for the Officers of the Society for the year 1914 and for and against the holding of a banquet in that year were counted.

The following officers were elected:

*President*, Rear-Admiral John R. Edwards, U. S. Navy.

*Secretary-Treasurer*, Lieutenant Commander Henry C. Dinger, U. S. Navy.

*Council:* Engineer-in-Chief R. S. Griffin, U. S. Navy.; Captain B. C. Bryan, U. S. Navy.; Commander S. S. Robison, U. S. Navy.

The vote on the banquet resulted as follows:

| | |
|---|---|
| For, | 114 |
| Against, | 85 |

The following is the financial statement of the Society for the year ending December 31, 1913:

### FINANCIAL STATEMENT, 1913.

The following statement shows the returns and costs involved in the issue of the JOURNAL of the Society for the year 1913:

*Returns.*

| | |
|---|---|
| Dues | $3,707.20 |
| Subscriptions | 1,824.40 |
| Sales | 515.99 |
| Advertisements | 1,822.50 |
| Exchange | 2.20 |

*Costs.*

| | | |
|---|---|---|
| Printing | $3,561.11 | |
| Engraving | 476.14 | |
| Drafting | 26.50 | |
| Manuscript | 559.45 | |
| Postage and expressage | 120.69 | |
| Commission | 136.39 | |
| Gross profit | 2,992.01 | |
| | $7,872.29 | $7,872.29 |

GENERAL PROFIT AND LOSS STATEMENT.

| | *Loss.* | *Profit.* |
|---|---|---|
| Gross profit on publication | | $2,992.01 |
| Interest | | 336.64 |
| Current profit and loss | | 6.63 |
| Salaries | $1,380.00 | |
| General expense | 53.30 | |
| Banquet, 1913 | 458.91 | |
| Net profit | 1,443.07 | |
| | $3,335.28 | $3,335.28 |

BALANCE STATEMENT.

| | | |
|---|---|---|
| Total assets January 1, 1913 | | $9,595.50 |
| *Assets December 31, 1913.* | | |
| Cash balance | $2,558.95 | |
| Accounts receivable, advertisements | 1,224.49 | |
| Accounts receivable, dues | 575.00 | |
| Accounts receivable, sales | 56.45 | |
| Accounts receivable, subscriptions | 354.43 | |
| Army and Navy Club bond (cost) | 1,000.00 | |
| Washington Railway and Electric bonds (cost) | 5,215.50 | |
| Furniture | 53.75 | |
| Net gain | | 1,443.07 |
| | $11,038.57 | $11,038.57 |

Market value of Washington Railway and Electric bonds carried in the above statement at cost was on December 31, 1913, 80. Total, $4,800.

Army and Navy Club bond, carried above at cost, is not a listed bond.

Audited and found correct.

H. C. DINGER, Lieut. Commander, U. S. N.
R. L. IRVINE, Lieut., U. S. N.
GEO. B. WRIGHT, Lieut., U. S. N.

The numerical strength of the Society on December 31, 1913, was—

| | |
|---|---|
| Members, | 414 |
| Associates, | 339 |
| Subscribers, | 363 |
| Exchanges, | 74 |
| | 1,190 |

## NOTICE OF ANNUAL BANQUET.

At a meeting of the Council it was decided to hold the Society's banquet on April 25th. It was further decided to make this banquet an occasion for bringing together and in closer touch with the service the engineering talent engaged in the production and development of naval material and thereby create closer mutual relations and opportunity for coöperation and exchange of ideas in sympathy with the aims and purposes of the Society.

It is proposed to invite representative men of the engineering field to discuss questions of general interest relating to naval engineering matters, and especially the matter of efficiently developing the engineering capabilities of the country as a Reserve Force of Naval Material.

The following committee to have charge of the arrangements and details of the banquet has been appointed:

Rear Admiral JOHN R. EDWARDS, U. S. N.
Captain GUSTAV KAEMMERLING, U. S. N.
Captain C. A. MCALLISTER, U. S. R. C. S.
Lieutenant Commander H. C. DINGER, U. S. N.
Lieutenant Commander J. B. GILMER, U. S. N.

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## SPECIAL NOTICE.

### *Material for Publication in "Journal."—Inducements for Securing Same.*

All members are urged to interest themselves in encouraging the submission of well-prepared papers on matters of live interest to the Naval Engineering Profession.

Suggestions as to the character of material that is desired to have published are also cordially invited.

The Society looks to its various members for assistance and coöperation in carrying out the aims of the Society. Suggestions for improvement of methods of conducting the business of the Society are also cordially invited.

It is felt that if all the units of the membership make it a point to do all they can toward the common cause of the Society, improved results will surely follow.

With a view of initiating a greater interest on the part of members to submit papers for publication in the JOURNAL the Council has decided to pay higher rates for papers specially suitable to its requirements.

It has been the practice to pay for articles at a minimum rate of a dollar a page. It is proposed now to pay more liberally and to increase this rate for specially good material or material which would involve considerable labor in its preparation. It is expected that the average rate paid will approximate to two dollars per page.

The following points should be considered in the preparation of papers submitted for publication:

Is the matter of interest to the membership of the Society?

Is it novel?

Does it convey information that will be of use?

Are the statements set forth accurate and in conformity with the best standards of engineering knowledge?

Notes and comments on matters of operation and description of methods of repair are desired as well as descriptions and tests of new apparatus.

Anything conveying valuable, new and interesting information concerning naval engineering matters is specially desired.

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THE FOLLOWING MEMBERS AND ASSOCIATES have joined the Society since the publication of the last JOURNAL:

MEMBERS.

Alden, Charles S., Ensign, U. S. N.
Bennett, Robert H., Ensign, U. S. N.
Cochrane, William F., Jr., Lieutenant, U. S. N.
Cooley, Hollis M., Lieutenant, U. S. N.
Craven, Francis S., Ensign, U. S. N.
Cummings, Damon E., Lieutenant, U. S. N.

Daniell, Jere R., Electric Boat Company, Groton, Conn.
Ford, Louis R., Ensign, U. S. N.
Foy, Edward J., Lieutenant, U. S. N.
Fuller, Henry G., Lieutenant, U. S. N.
Hall Wolcott E., Lieutenant, U. S. N.
Hinkamp, Clarence N., Lieutenant, U. S. N.
Keller, Harold R., Ensign, U. S. N.
King, Frank R., Lieutenant, U. S. N.
Lowe, Robert V., Lieutenant, U. S. N.
McCandlish, Benjamin V., Ensign, U. S. N.
McDowell, Clyde S., Lieutenant, U. S. N.
Mathews, James E., Bethlehem Steel Co., South Bethlehem, Pa.
Mayo, Claude B., Lieutenant, U. S. N.
Munroe, William R., Lieutenant, U. S. N.
Parker, James, Jr., Lieutenant, U. S. N.
Schelling, John M., Lieutenant, U. S. N.
Shafroth, John F., Jr., Lieutenant, U. S. N.
Soule, Charles C., Lieutenant, U. S. N.
Wickham, William C., Ensign, U. S. N.
Windsor, Charles C., Lieutenant, U. S. N.
Wolleson, Edwin A., Lieutenant, U. S. N.

ASSOCIATES.

Bonner, James B., 1514 Pennsylvania Building, Philadelphia, Pa.
Maas, Elov F., 650 Fourth Street, Bremerton, Wash.
Paterson, J. D., 8 Clairmont Gardens, Glasgow, Scotland.
Scott, Robert T., Chief Machinist, U. S. N.